PREFACE

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PREFACE

This book is the outcome of a series of lectures delivered to the Senior class of the Armour Institute of Technology, Chicago, Ill. It is primarily intended as a text-book for engineering students, but, it is hoped, will also be of interest to practicing engineers.

The field embraced by the title is a large one and it has been necessary to limit the treatment to essential elements. Much of the matter contained in the author's original notes, including that relating to steam engine design, valve gears, steam boiler design, and the like, has therefore been omitted. The numerous references appearing throughout the text and the appended bibliographies, which have been carefully compiled, are depended upon to extend the scope of the work. The standard codes of the American Society of Mechanical Engineers for conducting engine and boiler trials are in frequent demand by engineers and have therefore been included as an appendix.

Authorities have been freely consulted and extensive use made of current engineering literature, due acknowledgment being made by footnote or reference whenever possible.

The matter included is representative of American practice and no effort has been made to include any other except in a few special cases.

The author wishes to express his obligations to Prof. Raymond Burnham for many valuable suggestions and corrections, and to Mrs. Julia Beveridge, librarian at Armour Institute, for assistance in compiling references.

PREFACE TO SECOND EDITION

A NUMBER of additional changes have been made to bring this work into accord with more recent practice. All the typographical and other errors discovered in the first edition have been corrected.

PREFACE TO THIRD EDITION

ALL obsolete matter has been discarded, considerable new material has been added throughout the book, and many of the chapters have been entirely rewritten.

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STEAM POWER PLANT ENGINEERING

CHAPTER I.

ELEMENTARY STEAM POWER PLANTS.

1. General. — An equipment for the generation of power is known as a *station* or *plant*. When equipped to generate electricity for the production of light or power it is known as an *Electrical Station* or *Electric Light and Power Station*. The term *Heating Plant* refers to a plant in which the heat energy of fuel is made available for heating purposes through the medium of steam or hot water. In general, plants or stations are designated according to the manner in which the energy of the fuel is utilized.

When a station distributes power to a number of consumers more or less distant, it is called a *Central Station*. When the distances are very great, electrical current of high tension is frequently employed, and is transformed and distributed at convenient points through *Substations*. A plant designed to furnish power or heat to a building or a group of buildings under one management is called an *Isolated Station*. For example, the power plant of an office building is usually called an isolated station.

When the exhaust steam from the engines is discharged at approximately atmospheric pressure, the plant is said to be operating *non-condensing*. When the exhaust steam is condensed, reducing the back pressure on the piston by the partial vacuum thus formed, the plant is said to operate *condensing*.

When the exhaust steam may be used for manufacturing, heating, or other useful purposes, as is frequently the case in various manufacturing establishments, and in large office buildings, it is usually more economical to run non-condensing, while power plants for electric lighting and power, pumping stations, air compressor plants, and others, in which the load is fairly constant and the exhaust steam is not required for heating, are generally operated condensing.

2. Elementary Non-Condensing Plant. — Fig. 1 gives a diagrammatic outline of the essential elements of the simplest form of steam power plant. The equipment is complete in every respect and embodies all the accessories necessary for successful operation. Where a small

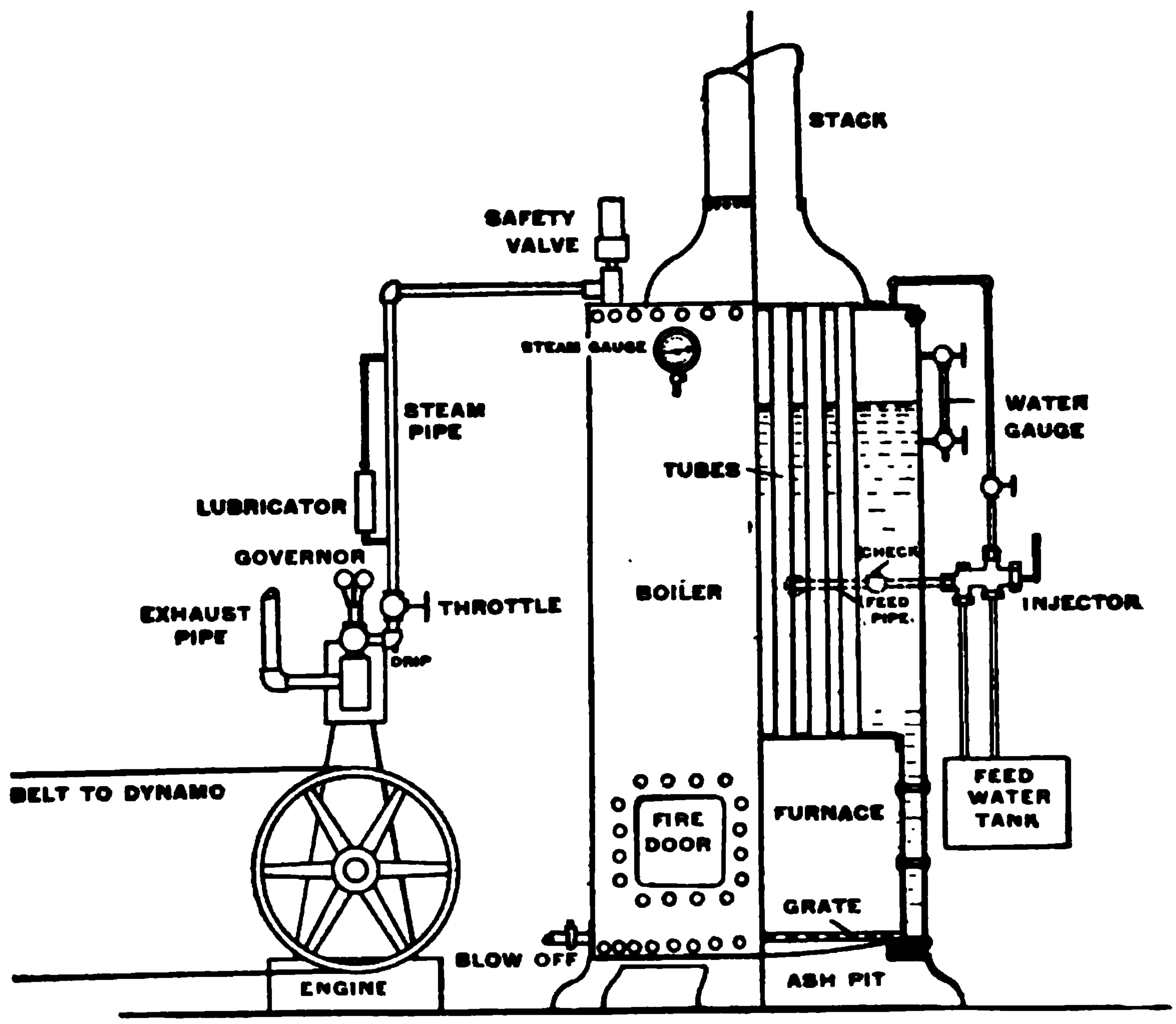


Fig. 1. Elementary Non-Condensing Plant.

amount of power is desired at intermittent periods, as in hoisting systems, threshing outfits and traction machinery, the arrangement is substantially as illustrated. The output in these cases seldom exceeds 50 horse power and the time of operation is usually short, so the cheapest of appliances are installed, simplicity and low first cost being more important than economy of fuel.

Such a plant has three essential elements: (1) The furnace, (2) the boiler, and (3) the engine. Fuel is fed into the *furnace*, where it is burned. A portion of the heat liberated from the fuel by combustion is absorbed by the water in the *boiler*, converting it into steam under pressure. The steam being admitted to the cylinder of the *engine* does work upon the piston, and is then exhausted through a suitable pipe to the atmosphere. The process is a continuous one, fuel and water being fed into the furnace and the boiler in proportion to the power demanded.

In such an elementary plant, certain *accessories* are necessary for successful operation. The *grate* for supporting the fuel during combustion consists of a cast iron grid or of a number of cast iron bars spaced in such a manner as to permit the passage of air through the fuel from below. The solid waste products fall through or are " sliced " through the grate bars into the *ash pit*, from which they may be removed through the *ash door*. The latter acts also as a means of regulating the supply of air below the grate. Fuel is fed into the furnace through the *fire door*, and when occasion demands, air may be supplied above the bed of fuel by means of this door. The *combustion chamber* is the space between the bed of fuel and the boiler heating surface, its office being to afford a space for the oxidation of the combustible gases from the solid fuel before they are cooled below ignition temperature by the comparatively cool surfaces of the boiler. The *chimney* or *stack* discharges the products of combustion into the atmosphere, and serves to create the draft necessary to draw the air through the bed of fuel. Various forced draft appliances are sometimes used to assist or to entirely replace the chimney. The *heating surface* is that portion of the boiler area which comes into contact with the hot furnace gases, absorbs the heat and transmits it to the water. In the small plant, illustrated in Fig. 1, the major portion of the heating surface is composed of a number of fire tubes below the water line, through which the heated gases pass. The volume above the water level is called the *steam space*. Water is forced into the boilers either by a *feed pump* or an *injector*. In small plants of the type considered, steam pumps are seldom employed; the injector answers the purpose and is considerably cheaper. A *safety valve* connected to the steam space of the boiler automatically permits steam to escape to the atmosphere if an excessive pressure is reached. The water level is indicated by *try cocks* or by a *gauge glass*, the top of which is connected with the steam space and the bottom with the water space. *Try cocks* are small valves placed in the water column or boiler shell, one at normal water level, one above it, and one below. By opening the valves from time to time the water level is approximately ascertained. They are ordinarily used in case of accident to the gauge glass. *Fusible plugs* are frequently inserted in the boiler shell at the lowest permissible water level. They are composed of an alloy having a low fusing point which melts when in contact with steam, thus giving warning by the blast of the escaping steam if the water level gets dangerously low. The *blowoff cock* is a valve fitted to the lowest part of the boiler to drain it of water or to discharge the sediment which deposits in the bottom. The steam outlet of a boiler is usually called the *steam nozzle*.

The essential accessories of the simple steam engine include: A *throttle valve* for controlling the supply of steam to the engine; the *governor*, which regulates the speed of the engine by governing the steam supply; the *lubricator*, attached to the steam pipe, which is usually of the "sight feed" class and provides for lubrication of piston and valve. Lubrication of the various bearings is effected by *oil cups* suitably located. *Drips* are placed wherever a water pocket is apt to form in order that the condensation may be drained. The apparatus to be driven by the engine may be *direct connected* to the crank shaft or *belted* to the *fly wheel* or *geared*.

In small plants of this type no attempt is made to utilize the exhaust steam except in instances where the stack is too short to create the necessary draft, in which case the exhaust may be discharged up the stack. If the draft is produced by convection of the heated gases in the chimney, the fuel is said to be burned under *natural draft;* if the natural draft is assisted by the exhaust steam, the fuel is said to be burned under *forced draft.* The power realized from a given weight of fuel is very low and seldom exceeds 2½ per cent of the heat value of the fuel. The distribution of the various losses in a plant of, say, 40 horse power is approximately as follows:

	B.T.U.
Heat value of 1 pound of coal	14,500
Boiler and furnace losses, 50 per cent	7,250
Heat of the steam, 50 per cent.	7,250
Heat equivalent of one horse power hour	2,545
Heat used to develop one horse power hour (50 pounds steam per horse power hour, pressure 80 pounds gauge, feed water 62 degrees F.)	57,500
	Per cent.
Percentage of heat in the steam, realized as work, $\frac{2,545}{57,500}$	4.4
Percentage of heat value of the coal realized as work, $\frac{2,545}{57,500 \div 0.50}$	2.2

The power plant of the modern locomotive is very much like that illustrated in Fig. 1, the main difference lying in the type of boiler and engine. The entire exhaust from the engine is discharged up the stack through a suitable nozzle, since the extreme rate of combustion requires an intense draft. The engine is a highly efficient one compared to that in the illustration, and the performance of the boiler is more economical. In average locomotive practice about 6 per cent of the heat value of the fuel is converted into mechanical energy at the draw bar. In general, a non-condensing steam plant in which the heat of the exhaust is wasted is very uneconomical of fuel, even under the

most favorable conditions, and seldom transforms as much as 7 per cent of the heat value of the fuel into mechanical energy.

3. Non-Condensing Plant. Exhaust Steam Heating. — Fig. 2 gives a diagrammatic arrangement of a simple non-condensing plant differing from Fig. 1 in that the exhaust steam is used for heating purposes. This shows the essential elements and accessories, but omits a number of small valves, by-passes, drains, and the like for the sake of simplicity. The plant is assumed to be of sufficient size to warrant the installation of efficient appliances. Steam is led from the boiler to the engine by the *steam main*. The moisture is removed from the steam before it enters the cylinder by a *steam separator*. The moisture drained from the separator is either discharged to waste or returned to the boiler. The exhaust steam from the engine is discharged into the *exhaust main* where it mingles with the steam exhausted from the steam pumps. Since the exhaust from engines and pumps contains a large portion of the cylinder oil introduced into the live steam for lubricating purposes, it passes through an *oil separator* before entering the heating system. After leaving the oil separator the exhaust steam is diverted into two paths, part of it entering the *feed water heater* where it condenses and gives up heat to the feed water, and the remainder flowing to the heating system. During warm weather the engine generally exhausts more steam than is necessary for heating purposes, in which case the surplus steam is automatically discharged to the *exhaust head* through the *back pressure valve*. The back pressure valve is, virtually, a large weighted check valve which remains closed when the pressure in the heating system is below a certain prescribed amount but which opens automatically when the pressure is greater than this amount. During cold weather it often happens that the engine exhaust is insufficient to supply the heating system, the radiators condensing the steam more rapidly than it can be supplied. In this case live steam from the boiler is automatically fed into the main heating supply pipe through the *reducing valve*.

The condensed steam, and the entrained air which is always present, are automatically discharged from the radiators by a *thermostatic valve* into the *returns header*. The thermostatic valve is so constructed that when in contact with the comparatively cool water of condensation it remains open and when in contact with steam it closes. The *vacuum pump* or vapor pump exhausts the condensed steam and air from the returns header and discharges them to the *returns tank*. The small pipe S admits cold water to the vacuum pump and serves to condense the heated vapor, and at the same time supply the necessary *make up water* to the system. The returns tank is open to the atmosphere so

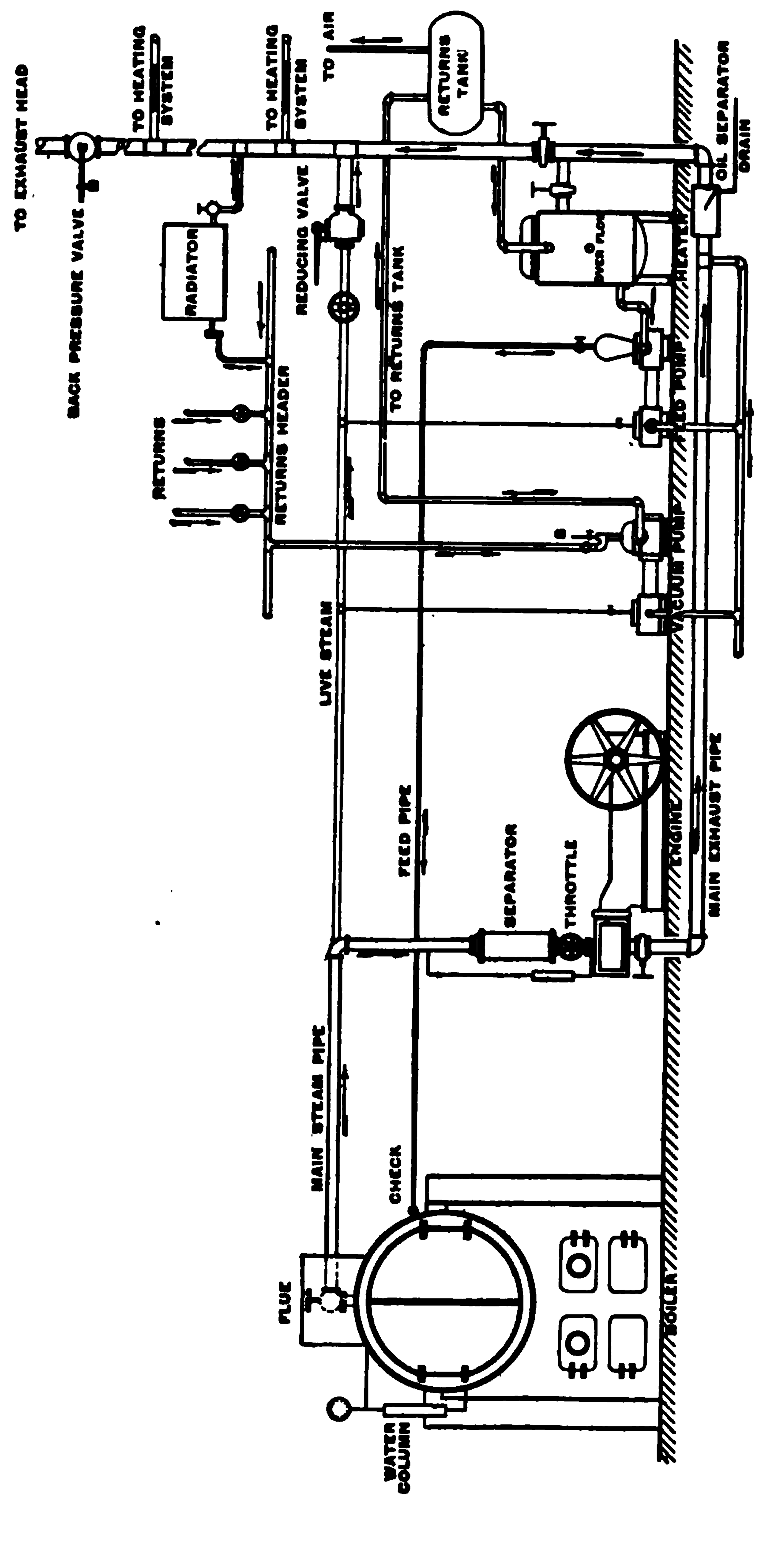


FIG. 2. Elementary Non-Condensing Plant with Heating System.

that the air discharged from the vacuum pump may escape. From the returns tank the condensed steam gravitates to the *feed water heater* where its temperature is raised to practically that of the exhaust steam. The feed water gravitates to the *feed pump* and is forced into the boiler. There are several systems of exhaust steam heating in current practice which differ considerably in details, but, in a broad sense, are similar to the one just described. The more important of these will be described later on.

During the summer months when the heating system is shut down, the plant operates as a simple non-condensing station and practically all of the exhaust steam, amounting to perhaps 60 per cent of the heat value of the fuel, is wasted. The total coal consumption, therefore, is charged against the power developed. During the winter months, however, all, or nearly all of the exhaust steam may be used for heating purposes and the power becomes a relatively small percentage of the total fuel energy utilized. The percentage of heat value of the fuel chargeable to power depends upon the size of the plant, the number and character of engines and boilers, and the conditions of operation. It ranges anywhere from 50 per cent to 100 per cent for the summer months and may run as low as 6 per cent for the winter months. This is on the assumption, of course, that the engine is debited only with the difference between the coal necessary to produce the heat entering the cylinder and that utilized in the heating system.

4. Elementary Condensing Plant. — Under the most favorable conditions a non-condensing plant can never be expected to realize more than 7 per cent of the heat value of the fuel as power. In large non-condensing power stations the demand for exhaust steam is usually limited to the heating of the feed water, and as only 12 per cent or 15 per cent can be utilized in this manner, the greater portion of the heat in the exhaust is lost. Non-condensing engines require from 20 to 60 pounds of steam per hour for each horse power developed. On the other hand in condensing engines the steam consumption may be reduced to as low as 10 pounds per horse power hour. The saving of fuel is at once apparent.

Fig. 3 gives a diagrammatic arrangement of a simple condensing plant in which the back pressure on the engine is reduced by condensing the exhaust steam. A different type of boiler from that in Fig. 1 or Fig. 2 has been selected for the purpose of bringing out a few of the characteristic elements. The products of combustion instead of passing directly through *fire tubes* to the stack as in Fig. 1 are deflected back and forth across a number of *water tubes*, by the *bridge wall* and a series of *baffles*. After imparting the greater part of their heat to the

FIG. 3. Simple Condensing Plant.

heating surface the products of combustion escape to the chimney through the *breeching* or *flue*. The rate of flow is regulated by a *damper* placed in the breeching as indicated.

The steam generated in the boiler is led to the engine through the *main header*. The steam is exhausted into a *condenser* in which its latent heat is absorbed by *injection* or cooling water. The process condenses the steam and creates a partial vacuum. The condensed steam, injection water, and the air which is invariably present are withdrawn by an *air pump* and discharged to the *hot well*. In case the vacuum should fail as by stoppage of the air pump the exhaust steam is automatically discharged to the *exhaust head* by the *atmospheric relief valve*, and the engine will operate non-condensing. The *atmospheric relief valve* is a large check valve which is held closed by atmospheric pressure as long as there is a vacuum in the condenser. When the vacuum fails the pressure of the exhaust becomes greater than that of the atmosphere and the valve opens.

The feed water may be taken from the hot well or from any other source of supply and forced into the *heater*. In this particular case it is taken from a cold supply and upon entering the heater is heated by the exhaust steam from the *air* and *feed pumps*. From the heater it gravitates to the feed pump and is forced into the boiler. Various other combinations of heaters, pumps, and condensers are necessary in many cases, depending upon the conditions of operation. Feed pumps, air pumps, and in fact all small engines used in connection with a steam power plant are usually called *auxiliaries*.

A well-designed station similar to the one illustrated in Fig. 3 is capable of converting about 10 per cent of the heat value of the fuel into mechanical energy. The various heat losses are approximately as follows:

BOILER LOSSES.	Per Cent.
Loss due to fuel falling through the grate	2
Loss due to incomplete combustion	2
Loss due to heat carried away in chimney gases	23
Radiation and other losses	8
Total	35

	B.T.U.
Heat used by engines and auxiliaries (16 pounds of steam per I.H.P. hour, pressure 150 pounds, feed water 210° F.)	16,250
Engine and generator friction, 5 per cent	812
Leakage, radiation, etc., 2 per cent	325
Total	17,387
Heat equivalent of one electrical horse power	2,545

	Per Cent.
Percentage of the heat value of the steam converted into electrical energy	14.7
Percentage of heat value of fuel converted into electrical energy $\frac{2545 \times 0.65}{17,387}$	9.5

5. Condensing Plant with Full Complement of Heat-saving Appliances. — When fuel is costly it frequently becomes necessary for the sake of economy to reduce the heat wastes as much as possible. The chimney gases, which in average practice are discharged at a temperature between 450 degrees F. and 550 degrees F., represent a loss of 20 per cent to 30 per cent of the total value of the fuel. If part of the heat could be reclaimed without impairing the draft the gain would be directly proportional to the reduction in temperature of the gases. Again, in some types of condensers all of the steam exhausted by the engine is condensed by the circulating water and discharged to waste. If provision could be made for utilizing part of the exhaust steam for feed-water heating, the efficiency of the plant could be correspondingly increased. In many cases the cost of installing such heat-saving devices would more than offset the gain effected, but occasions arise where they give marked economy.

Fig. 4 gives a diagrammatic arrangement of a condensing plant in which a number of heat-reclaiming devices are installed. The plant is assumed to consist of a number of engines, boilers, and auxiliaries. Coal is automatically transferred from the cars *to coal hoppers* placed above the boiler, by a system of buckets and conveyors. These hoppers store the coal in sufficient quantities to keep the boiler in continuous operation for some time. From the hoppers the coal is fed intermittently to the *stoker* by means of a *down spout.* The stoker feeds the furnace in proportion to the power demanded and automatically rejects the ash and refuse to the *ash pit.* The ashes are removed from the ash pit when occasion demands, and are transferred to the *ash hopper* by the same system of buckets and conveyor which handles the coal. The ash hopper is usually placed alongside the coal hoppers and is not unlike them in general appearance and construction.

The products of combustion are discharged to the stack through the flue or breeching. Within the flue is placed a feed-water heater called an *economizer*, the function of which is to absorb part of the heat from the gases on their way to the chimney. The heat reclaimed by the economizer varies widely with the conditions of operation and ranges between 5 per cent and 20 per cent. Since the economizer acts as a resistance to the passage of the products of combustion it is sometimes necessary to increase the draft either by increasing the height of the chimney or, as is the usual practice, by using a forced draft system.

Part of the heat of the exhaust steam is reclaimed by a *vacuum heater* which is placed in the exhaust line between engine and condenser. For example, if the feed water has a normal temperature of 60 degrees F. and the vacuum in the condenser is 26 inches, the vacuum heater will

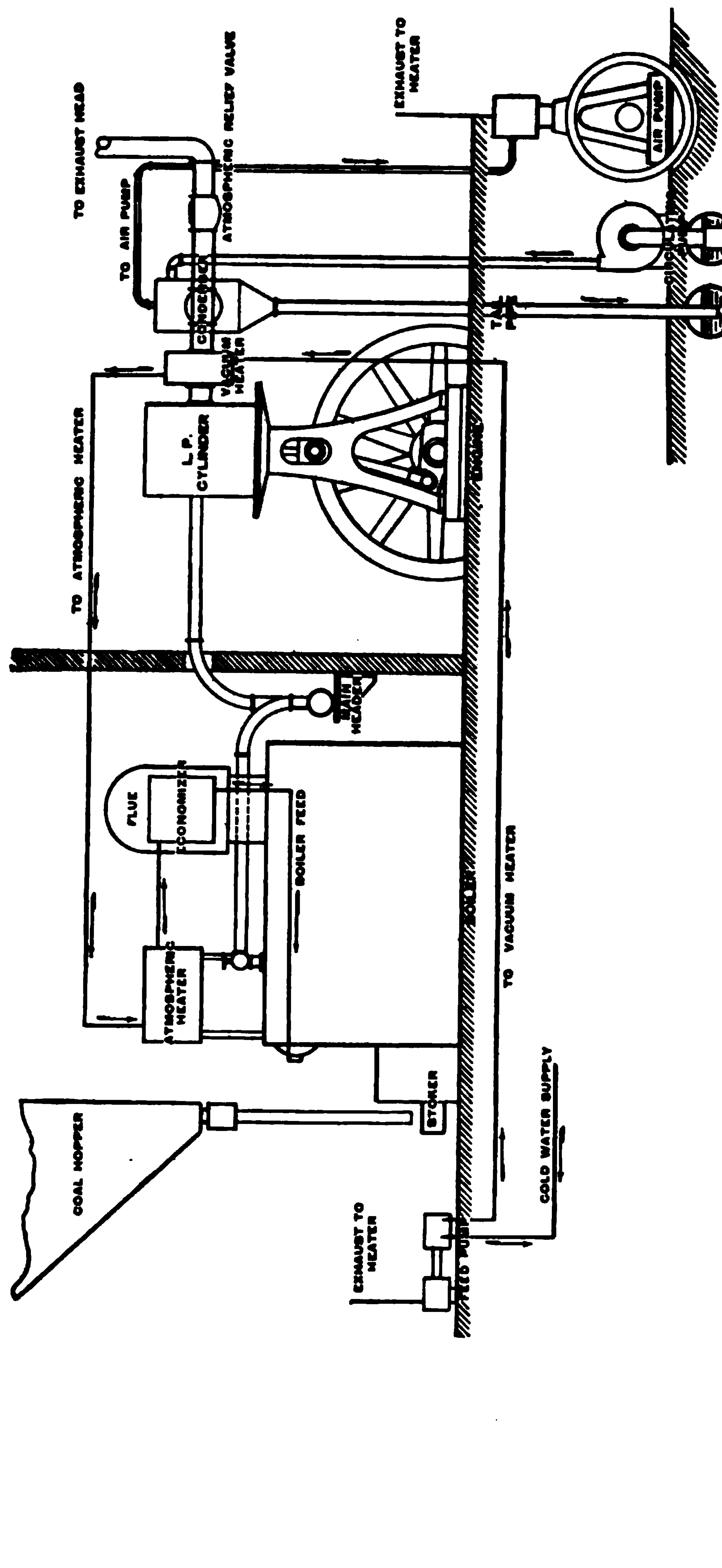


FIG. 4. Condensing Plant with Full Complement of Heat Saving Appliances.

raise the temperature of the feed to say 120 degrees F., thereby effecting a gain in heat of approximately 6 per cent. If the feed supply is taken from the *hot well* the vacuum heater is without purpose, as the temperature of the hot well will not be far from 120 degrees F.

Referring to the diagram, the path of the steam is as follows: From the boiler it flows through the *boiler lead* to the *main header* or equalizing pipe. From the main header it flows through the *engine lead* to the high-pressure cylinder. The exhaust steam discharges from the low-pressure cylinder through the vacuum heater and into the condenser. Part of the exhaust steam is condensed in the vacuum heater and gives up its latent heat to the feed water. The remainder is condensed by the injection water which is forced into the condenser chamber by the *circulating pump.* The condensed steam and circulating water gravitate through the *tail pipe* to the hot well. The air which enters the condenser either as leakage or entrainment is withdrawn by the air pump. The steam exhausted by the feed pump, air pump, stoker engine, and other steam-driven auxiliaries is usually discharged into the *atmospheric heater,* which still further heats the feed water.

Referring to the feed water, the circuit is as follows: The pump draws in cold water at a temperature of say 60 degrees F., and forces it in turn through the vacuum heater, the atmospheric heater, and the economizer into the boiler. The vacuum heater raises the temperature to 120 degrees F., the atmospheric heater increases it to 212 degrees F., and the economizer still further to about 300 degrees F. The heat reclaimed by this series of heaters is evidently the equivalent of that necessary to raise the feed water from 60 degrees F. to 300 degrees F., or approximately 24 per cent of the total heat supplied. In some plants the economizer only is installed, in others the economizer and atmospheric heater are deemed desirable, still others utilize all three. The distribution of the heat losses in a plant of this type operating under favorable conditions is approximately as follows:

	Per Cent.	B.T.U.
Delivered to engine, 15 pounds steam per I.H.P. hour; pressure 150 pounds, feed 60° F.	100	17,482
Delivered to feed pump	1.5	262
Delivered to circulating pump	1.5	262
Delivered to air pump	2	349
Delivered to small auxiliaries	1.5	262
Loss in leakage and drips	0.5	87
Engine and generator friction	5	874
Radiation and minor losses	1	175
Total		19,753

	Per Cent.	B.T.U.
Returned by vacuum heater	5.5	1,086
Returned by atmospheric heater	7.9	1,560
Returned by economizer	9.7	1,916
Total	23.1	4,562
Net heat delivered to engine in the form of steam to produce one electrical horse power, 19,753 − 4,562		15,191
Percentage converted to electrical power $\frac{2545}{15{,}191}$	16.7	
Boiler efficiency	70	
Percentage of heat value of fuel necessary to produce one electrical horse power at switchboard $\frac{2545 \times 0.70}{15{,}191}$	11.7	

The preceding figures give the results of very good practice. So much depends upon the size and character of the prime movers, the nature of the fuel, and the conditions of operation that no definite figure can be given for the percentage of heat converted to power in a given type of station. Six per cent represents good average practice in a non-condensing plant and 10 per cent in a condensing plant. Pumping stations operating continuously under full load have realized as much as 15 per cent of the total heat value of the fuel, but such performances are practically unobtainable in connection with steam-driven electrical power plants. Steam power plants as a class are very wasteful of fuel at the best.

One of the best recorded performances to date (March, 1909) of a steam-electric power plant is that of the Pacific Light and Power Company at Redondo, Cal. When operating under regular commercial conditions approximately 14 per cent of the available heat of the fuel (crude oil) is realized as power at the switchboard. For a detailed description of the plant and the results of the acceptance tests, see Jour. of Elec. Gas and Power, Aug. 22, 1908.

CHAPTER II.

FUELS AND COMBUSTION.

6. General. — The subject of fuels and combustion has been so extensively treated by various authorities that a comprehensive discussion would be without purpose here, but in order to bring out more clearly the matter pertaining to the commercial design and operation of steam power plants a few of the essential elements will be briefly treated.

The fuels used for steam making are coal, coke, wood, peat, mineral oil, natural and artificial gases, refuse products such as straw, manure, sawdust, tan bark, bagasse, and occasionally corn and molasses.

In most cases that fuel is selected which develops the required power at the lowest cost, taking into consideration all of the circumstances that may affect its use. Occasionally the disposition of waste products is a factor in the choice, but such instances are uncommon. The boilers and furnaces are designed to suit the fuel selected.

7. Classification of Fuels. — Fuels may be divided into three classes as follows:

1. Solid fuels.
   a. Natural fuels: straw, wood, peat, coal.
   b. Prepared: charcoal, coke, peat and other briquettes.
2. Liquid fuels.
   a. Natural: crude oils.
   b. Prepared: distilled oils, alcohol, molasses.
3. Gaseous fuels.
   a. Natural: natural gas.
   b. Prepared: coal gas, water gas, producer gas, oil gas.

8. Solid Fuels. — Solid fuels are of vegetable origin and exist in a variety of forms between that of a comparatively recent cellulose growth and that of nearly pure carbon as anthracite coal. They owe their forms to the conditions under which they were created or to the geological changes which they have undergone. With each succeeding stage the percentage of carbon increases. The chemical changes are approximately as follows:

Substance.	Carbon.	Hydrogen.	Oxygen.
	Per Cent	Per Cent	Per Cent
Pure cellulose	44.44	6.17	49.39
Wood	52.65	5.25	42.10
Peat	59.57	5.96	34.47
Lignite	66.04	5.27	28.69
Brown coal	73.18	5.58	21.14
Bituminous coal	75.06	5.84	19.10
Semi-bituminous coal	89.29	5.05	6.66
Anthracite	91.58	3.96	4.46
Graphite	100.00	...	...

All natural solid fuels contain more or less earthy or inorganic matter which is not combustible and therefore remains as ash, while the organic matter is consumed. Sometimes the percentage of ash is so great as to render them valueless for steam-making purposes.

Origin and Formation of Fuel: Engng, Aug. 23, 1901; Am. Geol., Feb., 1899; Col. Guard, Sept. 10, 1897, Oct. 1, 1897, Jan. 14, 1898, Jan. 28, 1898, March 18, 1898, Sept. 14, 1900; Ec. Geol., Oct., 1905; Eng. U.S., April 1, 1903; Ir. and Coal Td. Review, Feb. 4, 1898, July 13, 1906.

9. Coal.—Coals are most satisfactorily classified according to the constituents of the combustible, as

	Fixed Carbon.	Volatile Matter.
	Per Cent	Per Cent
Anthracite	97 to 92.5	3 to 7.5
Semi-anthracite	92.5 to 87.5	7.5 to 12.5
Semi-bituminous	87.5 to 75	12.5 to 25
Bituminous, Eastern	75 to 60	25 to 40
Bituminous, Western	65 to 50	35 to 50
Lignite	Under 50	Over 50

Classification of Coals: Am. Inst. of Min. Engrs., May, 1906, Sept., 1905; Mines and Minerals, Dec., 1906; Min. Rept., Apr. 26, 1906; Col. Guard, July 6, 1900; Power, Oct., 1906.

10. Anthracite. — This is the most perfect form of coal and consists almost entirely of carbon; it contains very little hydrocarbon and burns with little or no smoke, is slow to ignite, burns slowly, and breaks into small pieces when rapidly heated. It requires a very large grate of about twice the surface necessary for bituminous coal. Large sizes may be burned in almost any kind of a furnace and with moderate draft. For small sizes a thinner bed has to be carried unless a strong draft is used. There is difficulty in keeping it free from air holes. When

possible, the coal should be at least six inches deep on the grates. On account of the large percentage of ash in the smaller size, the fire requires frequent cleaning. Anthracite does not require "slicing" and should be disturbed only when cleaning is necessary.

Small Size Anthracite: Eng. and Min. Jour., Dec. 22, 1904. *Heat Value of Anthracite, Small Sizes, and the Best Way of burning it:* Col. Guard, Nov. 26, 1897. *Anthracite Coal Mines and Coal Mining:* Rev. of Rev., July, 1902. *The Screening of Anthracite:* Col. Guard, Sept. 20, 1901. *Preparation of Anthracite:* Mines and Min., March, 1905. *Anthracite Washeries:* Am. Inst. of Min. Engrs., Nov., 1905. *Anthracite Coal Fields of Pennsylvania:* Min. Mag., March, 1905. *Virginia Anthracite:* Eng. News, Oct. 20, 1904; Mines and Min., March, 1906. *Burning of Anthracite Culm of Poor Quality:* Trans. A.S.M.E., 7–390. *The Use of Electricity in Anthracite Mining:* Eng. and Min. Jour., Feb. 2, 1907.

Anthracite is classed and marketed according to sizes, the following division of mesh being adopted as standard at Wilkesbarre in 1891:

Egg	coal	must	pass	through	2¾	inch	mesh	and	not	through	2 inch
Stove	"	"	"	"	2	"	"	"	"	"	1¼ "
Chestnut	"	"	"	"	1¼	"	"	"	"	"	¾ "
Pea	"	"	"	"	¾	"	"	"	"	"	½ "
Buckwheat	"	"	"	"	½	"	"	"	"	"	¼ "
Rice	"	"	"	"	¼	"	"	"	"	"	⅛ "

Sizes over "pea coal" are prohibitive in price for steam power plant use and consequently the demand is limited to the smaller sizes.

11. Semi-Anthracite. — This coal kindles more readily and burns more rapidly than anthracite. It requires little attention, burns freely with a short flame, and yields great heat with little clinker and ash. It is apt to split up on burning and waste somewhat in falling through the grates. It swells considerably but does not cake. It has less density, hardness, and metallic luster than anthracite, and can generally be distinguished by its tendency to soil the hands while pure anthracite will not.

12. Semi-Bituminous. — This coal is softer than semi-anthracite. It ignites easily and burns freely under a moderate draft. It gives an intense fire and is an excellent steam coal, but is apt to smoke considerably unless special provision is made to prevent it.

13. Bituminous. — This coal contains a large and varying amount of volatile matter and requires careful firing to prevent smoke and clinker. Its physical properties vary widely, so much so that it is usually divided into three grades:

1. *Dry Bituminous* coal is sometimes known as the free-burning bituminous. It is hard and dense, black in color, but brittle and

splintery. It ignites somewhat slowly, burns freely with a short, clean, bluish flame, little smoke, and without caking.

2. *Bituminous Caking* coal swells up, becomes pasty, and fuses together in burning. It contains less fixed carbon and more volatile matter than the free-burning grades. Caking coal is rich in hydrocarbon and is particularly adapted to gas making. The flame is of a yellowish color.

3. *Long Flaming Bituminous* coal is similar in many respects to the caking coal but contains a larger percentage of volatile matter. It is free burning, with a long, yellowish flame. It may be either caking or non-caking.

COAL FIELDS OF THE UNITED STATES.

Alabama: Mines and Minerals, May, 1901.

Arkansas: Eng. and Min. Jour., Sept. 12, 1903, Oct. 28, 1905.

Colorado: Min. Rept., Jan. 19, 1905, March 2, 1905; Jour. W. S. Engrs., Dec., 1903; Mines and Minerals, May, 1905.

Illinois: Min. Mag., March, 1905; Eng. and Min. Jour., Jan. 13, 1906.

Indiana: Eng. Rec., Jan. 27, 1906; Min. Mag., March, 1905; Power, July and Aug., 1902.

Indian Territory: Min. Rept., May 17, 1906.

Kansas: Eng. and Min. Jour., Dec. 7, 1901.

Kentucky: Col. Guard, Sept. 7, 1900.

Michigan: Eng. and Min. Jour., June 30, 1900; Min. World, Feb. 9, 1907.

Missouri: Am. Inst. of Min. Engrs., Jan., 1905.

Montana: Min. Mag., March, 1905; Min. World, Nov. 24, 1906.

Ohio: Min. Mag., March, 1905.

Pennsylvania: Eng. and Min. Jour., Aug. 24, 1901; Trans. A.S.M.E., 4–217; Min. Mag., March, 1905; Pro. Eng. S. W. Penn., Jan., 1907.

Texas: Mines and Min., Oct., 1905.

Virginia: Mines and Min., March, 1906; Eng. News, Oct. 20, 1904.

West Virginia: Eng. and Min. Jour., May 12, 1904.

Wyoming: Min. World, May 6, 1905.

GENERAL.

Coal Mines of the United States: Peabody Atlas, A. Bement, Chicago, Ill., Min. World, May 6, 1905.

Coal Resources of the Pacific: Eng. Mag., May, 1902.

Rocky Mountain Coal Fields: Min. Rept., Jan. 5, 1905; Jour. Asso. Eng. Soc., Dec., 1902.

Coal Fields, U.S. Northwest: Rev. of Rev., Feb., 1903.

Coal Fields, U.S. Southwest: Eng. and Min. Jour., Oct. 17, 1903.

U.S. Coal Fields: Steam Boiler Practice, Wm. Kent.

Report of Coal Testing Plant: U.S. Geological Survey, Washington, D.C.

14. Lignite, or brown coal, is a substance of more recent geological formation than coal and represents a stage in development intermediate between coal and peat. Its specific gravity is low, 1.2, and when freshly mined it contains as high as 50 per cent of moisture. It is non-caking and gives a bright but slightly smoky flame. It is a low-grade fuel and is used where good coal is difficult to get. Exposure to the air causes it to split into fine pieces like air-slacked lime. It is very fragile and will not bear much handling in transportation.

Eng. and Min. Jour., Nov. 22, 1902, Feb. 7, 1903; Mines and Min., July, 1901. *Lignite of Northeastern Wyoming:* Mines and Min., Feb., 1907.

15. Peat, or Turf, is formed by the slow carbonization under water of a variety of accumulated vegetable materials. It is unsuitable for fuel until dried. Peat as ordinarily cut and dried is too bulky for commercial competition with coal, and is used only where coal is prohibitive in price. When properly prepared and compressed into briquettes peat is an excellent fuel. In Russia, Germany, and Holland peat briquettes have passed the experimental stage and several millions of pounds are manufactured annually. Peat is used but little in this country at present, but its possibilities are beginning to attract the attention of engineers. The proportion in which the various primary constituents exist in dried peat is approximately as follows:

	Per Cent.
Fixed carbon	35
Volatile matter	60
Ash	5

Peat: Power, Sept., 1907; Engr. U.S., April 1, 1905; Min. World, Sept. 30, 1905; Col. Guard, Nov. 30, 1900; Mines and Min., July, 1901; Eng. and Min. Jour., Nov. 22, 1902; Feb. 7, 1903, Eng. Rec., 52–191; Sci. Am. Sup., March 2, 1907; Elec. Engr., Lond., Dec. 6, 1907.

Fuel Briquetting: Jour. Assn. Eng. Soc., Jan., 1906; Iron Age, April 19, 1906; Power, Dec., 1902; Eng. and Min. Jour., Nov. 8, 1902; Mines and Min., Oct., 1904; Power, March, 1905; Engr. U.S., May, 1905.

16. Wood, Straw, Sawdust, Bagasse, Tanbark. — In certain localities cord wood is still used as a fuel, but the steadily increasing values of even the poorest qualities are rapidly prohibiting its use for steam-generating purposes. Sawdust, shavings, tanbark and other waste products of wood are burned under boilers in situations where such disposition nets the best financial returns. Recent progress, however, in industrial chemistry shows that ethyl and wood alcohols and other valuable by-products can be cheaply made from sawdust, shavings,

slashings and similar waste material, and it is not unlikely that their use for steaming purposes will be unheard of in a comparatively few years. Table 00 gives the physical and chemical characteristics of a number of woods.

Wood as Fuel: Prac. Engr. U. S., Jan., 1910, p. 805; Power & Engr., June 30, 1908, p. 1015; Power, Dec., 1908, p. 772.

Burning Sawdust: Prac. Engr. U. S., Jan., 1910, p. 48; Power & Engr., April 7, 1908, p. 536; Oct. 13, 1908, p. 613; Jour. of Elec., Oct., 1905.

TABLE 00.

PHYSICAL AND CHEMICAL PROPERTIES OF WOODS, STRAW AND TANBARK.

(Prac. Engr. U. S., Jan., 1910.)

	Weight per Cubic Foot. Pounds.	Weight per Cord. Pounds.	Equivalent Weight of Coal. 13,500 B.T.U.	Carbon. Per Cent.	Hydrogen. Per Cent.	Oxygen. Per Cent.	Nitrogen. Per Cent.	Ash. Per Cent.	Calorific value, B.T.U. per Pound.	Authority.
Ash	46	3520	1420	......	......	......	......	......	5450	Hutton
Beech	43	3250	1300	49.36	6.01	42.69	0.91	1.06	5400	Sharpless
Birch	45	2880	1190	50.20	6.20	41.62	1.15	0.81	5580	Hutton
Cherry	42	3140	1260	......	......	......	......	......	5420	"
Chestnut	41	2350	940	......	......	......	......	......	5400	Sharpless
Elm	35	2350	940	......	......	......	......	......	5400	"
Hemlock	25	1220	580	......	......	......	......	......	6410	Hutton
Hickory	53	4500	1800	......	......	......	......	......	5400	Sharpless
Maple, Hard	49	3310	1340	......	......	......	......	......	5460	Hutton
Oak, Live	59	3850	1560	......	......	......	......	......	5460	"
" White	52	3850	1540	49.64	5.92	41.16	1.29	1.97	5400	Rankine
" Red	45	3310	1340	......	......	......	......	......	5460	Hutton
Pine, White	25	1920	970	......	......	......	......	......	6830	"
" Yellow	36	2130	1050	......	......	......	......	......	6660	"
Poplar	36	2130	1050	49.37	6.21	41.60	0.96	1.86	6660	"
Spruce	25	1920	970	......	......	......	......	......	6830	"
Walnut	35	3310	1340	......	......	......	......	......	5460	"
Willow	25	1920	970	49.96	5.96	39.56	0.96	3.37	6830	Rankine
Average	....	.....	......	49.70	6.06	41.30	1.05	1.80		
Straw:	6 to 8 *		Water							
Wheat		.....	16.00	35.86	5.01	37.68	0.45	5.00	......	Clark
Barley		.....	15.50	36.27	5.07	38.26	0.40	4.50	......	"
Average			15.75	36.06	5.04	37.97	0.42	4.75	5155	
Tanbark Dry	....	.....	......	51.80	6.04	40.74	......	1.42	9500	Myers

* Compressed.

Bagasse, or megass, is refuse sugar cane and is used as a fuel on the sugar plantations. Its heat value depends upon the proportions of fiber, molasses, sugar and water left after the extraction. The heat furnished by the different constituents is about as follows: Fiber, 8325 B.T.U. per pound; sugar, 7223 B.T.U. per pound; and molasses, 6956 B.T.U. per pound. Table 0 gives the heat value of bagasse and

TABLE 0.

HEAT VALUES OF BAGASSE AND VARIATION WITH DEGREE OF EXTRACTION.

Per Cent Extraction on Weight of Cane	Per Cent Moisture in Bagasse.	Fiber.		Sugar.		Molasses.		Total Heat developed. B.T.U.	[illegible]	[illegible]	[illegible] Power.	Coal Equivalent per Ton of Cane. Pounds.	Temperature of Fire. Fahr.
		Per Cent in Bagasse.					Fuel Value, B.T.U.						
90	0.00	100.0(					....	8325				119	2465°
85	28 33	66.67					116	5900				119	2236
80	42.50	50.0(					174	[illegible]	509	4188	3.34	120	2023
75	51.00	40.0(					209	3972	611	3361	4.17	120	1862
70	56 67	33 33	2775	6.67	482	3.33	232	3489	679	2810	4.98	120	1732
65	60.71	28.57	2378	7.15	516	3.57	248	3142	727	2415	5.80	121	[illegible]
60	63.75	25.00	2081	7.50	541	3.75	261	2883	764	2119	6.61	121	1513
55	66.12	22 22	1850	7.78	562	3 88	270	[illegible]	792	1890	7.40	121	1427
50	68 00	20.00	1665	8.00	578	4 00	278	2521	815	1706	8 21	122	1350
45	69 55	18.18	1513	8 18	591	4 09	284	2388	833	1555	9.00	122	1284
40	70.83	16.67	1388	8 33	601	4.17	290	2279	849	1430	9.79	123	[illegible]
25	73 67	13.33	1110	8.67	626	4 33	301	2037	883	1154	12 13	124	1077
[illegible]	75.00	11.77	980	8.82	637	4.41	307	1924	899	1025	13.66	124	1002
0	76.50	10.00	832	9.00	650	4.50	313	[illegible]	916	879	15.93	126	906

variation with the degree of extraction. A typical furnace for burning bagasse is shown in Fig. 4a.

Bagasse as Fuel: Prac. Engr. U. S., Jan., 1910; Engr. U. S., April 1, 1903; Jour. Assn. Engng. Soc., July, 1901; Engng., Feb. 18, 1910.

Tanbark is usually quite moist; the amount of moisture varies with the leaching process used and averages around 65 per cent. In this condition it has a heat value of about 4300 B.T.U. per pound. If perfectly dry its heating power is approximately 6100 B.T.U. per pound. As in the case of all moist fuels, tanbark must be surrounded by heated surfaces of sufficient extent to insure drying out the fresh fuel as thrown on the fire. A successful furnace for burning tanbark is shown in Fig. 4b.

Tanbark as a Boiler Fuel: Jour. A.S.M.E., Feb., 1910, p. 181; Jour. A.S.M.E., Oct., 1909, p. 951; Prac. Engr. U. S., Jan., 1910.

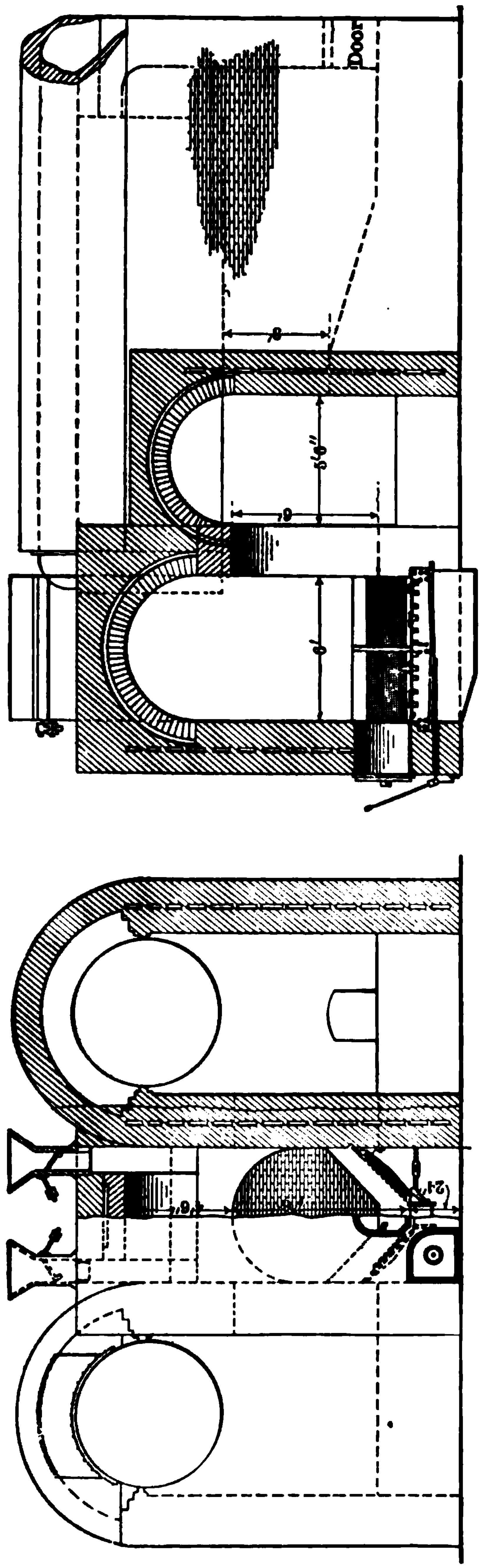


FIG. 4a. Sectional Elevations of the Myers Furnace for Burning Bagasse.

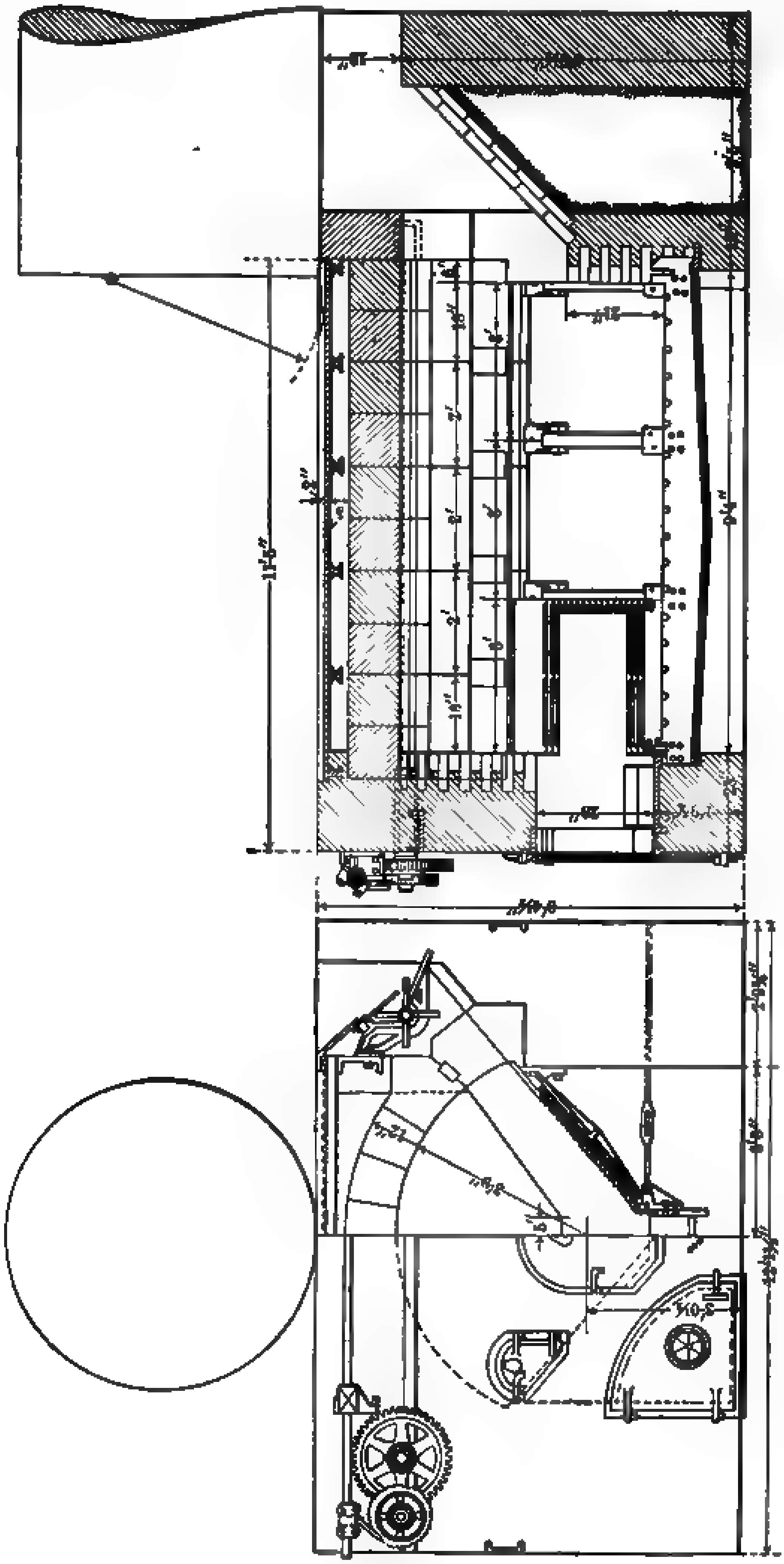


FIG. 4b. End and Side Sectional Elevation of the Myers Tanbark Furnace.

17. Composition of Coal. — The uncombined carbon in coal is known as *fixed carbon*, while the hydrocarbons and other gaseous compounds which distill off on application of heat constitute the *volatile matter*. Refractory earths and moisture are found in varying quantities in different classes of coal and as they are incombustible tend to reduce the heat value of the fuel. That part of the fuel which is dry and free from ash is called the *combustible*, though the nitrogen and oxygen in the volatile matter are not actually combustible. The term "pure coal" has been suggested in this connection and is meeting with much favor. (Jour. W.S.E. 11–757.)* The various elementary constituents of a fuel must be determined by a careful chemical analysis, but in most cases it is only necessary to know the heating value, the per cent of moisture and ash and perhaps the per cent of sulphur. Table 1 shows the composition of a number of American coals and gives a good idea of their chemical characteristics.

TABLE 1.

COMPOSITION OF TYPICAL AMERICAN COALS.

(U.S. Geological Survey.)

	Anthracite.			Semi-Bituminous.		
	Lehigh, Mine Run*.	Lykens Valley, Mine Run.*	Scranton, Culm.	Pocahontas, Mine Run.	George's Creek, Mine Run.*	Arkansas, Half Lump, Half Slack.
Proximate analysis						
Water	1.97	1.50	2.08	0.65	0.59	1.28
Volatile matter	4.35	7.84	7.27	18.80	18.52	12.82
Fixed carbon	86.49	81.07	74.32	75.92	74.31	73.69
Ash	7.19	9.59	16.33	4.63	6.58	12.21
Sulphur	0.64	0.50	0.77	0.57	0.81	2.01
Ultimate analysis						
Carbon	85.66	83.20	75.21	85.91	81.05	77.29
Hydrogen	2.78	3.29	2.81	4.58	4.91	3.74
Nitrogen	0.77	0.95	0.80	1.07	2.15	1.39
Oxygen	2.87	2.45	4.08	3.24	4.57	3.36
Calorific value						
Calorimeter	.....	.....	12472	15190	.....	13406
Dulong's Formula	13963	13954	12395	15104	14484	13831

* H. J. Williams.

* See also "Unit Coal and the Composition of Coal Ash," Univ. of Ill. Bulletin No. 37, Aug. 9, 1909.

TABLE 1.—*Continued.*

	Bituminous.				Lignite.	
	Big Muddy, Carterville Ill., Mine Run.	Jamison, Penn., Lump.	Straight Creek, Ky., Mine Run.	St. Claire County, Ill., Slack.	Tesla, Cal., Mine Run.	Gallup, New Mexico, Mine Run.
Proximate analysis						
Water	8.61	3.15	1.92	5.31	18.51	10.86
Volatile matter	32.40	30.27	36.56	34.29	35.33	35.14
Fixed carbon	51.33	56.17	57.08	36.24	30.67	46.90
Ash	7.66	10.41	4.44	24.16	15.49	7.10
Sulphur	1.65	1.26	1.24	4.30	3.05	0.64
Ultimate analysis						
Carbon	68.14	74.33	78.31	54.06	47.34	64.34
Hydrogen	5.38	4.96	5.36	4.57	5.93	5.73
Nitrogen	1.34	1.43	1.85	0.78	0.66	1.05
Oxygen	15.83	7.61	8.80	12.13	27.53	21.14
Calorific value						
Calorimeter	12236	13406	14319	9848	8525	11435
Dulong's Formula	12082	13371	14081	9929	8550	11299

Report of Committee on Coal Analysis: Jour. Amer. Chem. Soc., 21-1116, 1899. *Coal Testing:* Mines and Min., Nov., 1905. *Coal Testing:* Col. Guard, April 12, 1906. *Report of Government Coal-Testing Plant:* Trans. A.S.M.E., Dec., 1905; Eng. and Min. Jour., Sept. 15, 1904; Mines and Min., June, 1906. *Determination of Volatile Combustible Matter:* Jour. Am. Chem. Soc., 21-1137, 1899; *ibid.*, 28-1002, 1906. *Proximate Constituents of Coal:* Jour. Gas Light, Jan. 5, 1897. *Ultimate Analysis:* Col. Guard, Oct. 15, 1897. *Analysis of Coal:* Trans. A.S.M.E., 21-65. *Sampling Coal:* Chem. Engr., Nov., 1905; Am. Inst. Mech. Engrs., Sept., 1905. *Illinois Coal Tests:* Eng. News, Feb. 7, 1907.

Sulphur in Coal: Eng. and Min. Jour., June 27, 1903; Mines and Min., Feb., 1906; Jour. Am. Chem. Soc., 24-852, 25-184, 26-1139.

Ash: Relation between Composition and Fusibility in Coal Ash: Col. Guard, Oct., 1897, March 18, 1898. *Composition of Ash:* Stromeyer, Steam Boilers, p. 11.

18. Combustion. — By combustion is meant the chemical union of the combustible material of a fuel and the oxygen of the air. Theoretically the process is a simple one, as it is only necessary to bring each particle of fuel previously heated to the kindling temperature in contact with the correct amount of oxygen and the combustion will be complete, the fuel oxidizing to the highest possible degree. In practice, however, the size and character of fuel, type of furnace, draft, impurities in the fuel, and the mechanical difficulties affect combustion to such an extent as to render oxidation more or less incomplete.

When heat is applied to coal combustion takes place in three separate and distinct stages:

1. Absorption of heat. A fresh charge of fuel when thrown on a fire must first be brought to the kindling point in order that chemical

action may take place. The temperatures necessary to cause this union of oxygen and fuel are approximately as follows:

	Degrees F.		Degrees F.
Lignite dust	300	Cokes	800
Sulphur	470	Anthracite lump	750
Dried peat	435	Carbon monoxide	1211
Anthracite dust	570	Hydrogen	1100
Lump coal	600		

(Stromeyer, Marine Boiler Management and Construction, p. 93.)

2. Vaporization of the hydrocarbon portion of the fuel and its combustion, the hydrocarbons consisting principally of olefiant gas, C_2H_2, marsh gas, CH_4, tar, pitch, naphtha and the like. As these gases are driven off they become mixed with the entering air, and the carbon and hydrogen unite with the oxygen, forming carbon dioxide, CO_2, and water vapor, H_2O, respectively, and give up heat in doing so. If volatile sulphur is present it unites with the oxygen, forming sulphur dioxide, SO_2, and also gives up heat, but its presence is objectionable, as the SO_2, particularly, in the presence of moisture, attacks the metal of the furnace and boiler and causes rapid corrosion. If insufficient oxygen is present for complete oxidation, the carbon may burn to carbon monoxide, CO, and only a small portion of the available heat be liberated.

3. Combustion of the solid or carbonaceous portion of the fuel. After the hydrocarbons have been driven off and oxidized the remaining solid matter is composed chiefly of carbon and ash. The carbon unites with the oxygen, forming carbon dioxide, carbon monoxide, or both, depending upon the completeness of combustion. The ash, of course, remains unconsumed.

In commercial practice the requirements for perfect combustion are a surplus of air, a thorough mixture of the fuel particles with the air, and a high temperature. The surplus of air above theoretical requirements should be kept to a minimum, but even in the most scientifically designed furnace some excess is essential on account of the difficulty of properly mixing the gases, since the currents of combustible gases and air are apt to be more or less stratified. The products of combustion must be maintained at the kindling temperature until oxidation is complete, otherwise the carbon will be wasted as carbon monoxide or as smoke. The final products of combustion as exhausted by the chimney should consist only of carbon dioxide, water vapor, oxygen, nitrogen, and the oxides of impurities in the fuel.

When the combustible elements unite with oxygen they do so in definite proportions, called the *combining weights*, which are always the same, and the union produces a fixed quantity of heat. Thus in the

combustion of carbon to CO, 12 pounds of carbon unite with 16 pounds of oxygen, forming 28 pounds of CO; hence one pound of carbon will form

$$\frac{12+16}{12} = 2\tfrac{1}{3} \text{ pounds of CO.}$$

The heat of combustion will be 4,451 B.T.U. per pound of carbon thus consumed.

TABLE 2.

DATA RELATIVE TO ELEMENTS MOST COMMONLY MET WITH IN CONNECTION WITH COMBUSTION OF FUEL.

Substance.	Chemical Formula.	Combining Weight.	Chemical Reaction.	Oxygen. Weight per Pound Substance in First Column.	Air. Weight per Pound Substance in First Column.
Acetylene	C_2H_2	26	$C_2H_2+5\,O=2\,CO_2+H_2O$	3.08	13.27
Air	.....	.....	.....	.....	.....
Ash	.....	.....	.....	.....	.....
Carbon	C	12	$C+O=CO$	1.33	5.78
Carbon	C	12	$C+2\,O=CO_2$	2.66	11.58
Carbon dioxide	CO_2	44	.....	.....	.....
Carbon monoxide	CO	28	$CO+O=CO_2$	0.57	2.47
Hydrogen	H	1	$2H+O=H_2O$	8	34.8
Marsh gas	CH_4	16	$CH_4+4\,O=CO_2+2\,H_2O$	4	17.4
Nitrogen	N	14	.....	.....	.....
Olefiant gas	C_2H_4	28	$C_2H_4+6\,O=2CO_2+2\,H_2O$	3.43	14.9
Oxygen	O	16	.....	.....	.....
Sulphur	S	32	$S+2\,O=SO_2$	1	4.32
Water vapor	H_2O	18		.....	.....

Substance.	Specific Heat at Constant Pressure.*	Weight per Cubic Foot. 62° F.	Cubic Feet per Pound. 30″ Hg.‡	Heat of Combustion B.T.U.† Per Pound.	Heat of Combustion B.T.U.† Per Cubic Foot. 62° F. 30″ Hg.
Acetylene	.....	0.0685	14.6	21,465	1470
Air	0.2375	0.0759	13.16	.....	.....
Ash	0.2	.....	.....	.....	.....
Carbon	.....	.....	.....	4,451	.....
Carbon	.....	.....	.....	14,544	.....
Carbon dioxide	0.2169	0.1159	8.627	.....	.....
Carbon monoxide	0.2426	0.0737	13.55	4,325	320
Hydrogen	3.409	0.00527	189.8	62,032	326
Marsh gas	0.593	0.0421	23.72	23,513	991
Nitrogen	0.2438	0.07376	13.55	.....	.....
Olefiant gas	0.404	0.0737	13.55	21,344	1570
Oxygen	0.2175	0.0843	11.86	.....	.....
Sulphur	0.1544 §	0.1686	5.93	.....	.....
Water vapor	0.48	.....	.....	.....	.....

* Regnault. † Favre & Silberman. ‡ Smithsonian Tables. § SO_2.

Similarly in burning to CO_2 one pound of carbon will form $3\frac{2}{3}$ pounds of CO_2 and liberate 14,544 B.T.U.

Table 2 gives the physical and chemical properties of the substances most commonly met with in connection with combustion.

Combustion: Engr. U. S., Feb. 16, 1903, April 1, 1903, Jan. 1, 1907; Power, Feb. 1905, Oct. 26, 1906; Eng. Mag., July, 1907; Prac. Engr. U. S., Jan., 1910.

Simple Method of Determining Condition of Combustion: A. Bement, Jour. West. Soc. Eng., June, 1901. *Effect of Altitude on Combustion:* Power, Sept., 1906. *Notes on Fuel Combustion in Power Plants:* Elec. Engr., Lond., Aug. 10, 1906.

19. Temperature due to Combustion. — If the heat liberated by the chemical union of any two elements is confined in such a way that no radiation can take place, the resulting increase in temperature of the products of combustion may be expressed

$$t = \frac{h}{ws}, \tag{1}$$

in which

t = the increase in temperature, degrees F.
h = the heat liberated, B.T.U. per pound of combustible.
s = the specific heat of the resulting gaseous products.
w = weight of the gaseous products, pounds.

Thus 1 pound of carbon in burning to CO_2 combines with $2\frac{2}{3}$ pounds of oxygen and forms $3\frac{2}{3}$ pounds of CO_2. Taking the mean specific heat of CO_2 as 0.217, the resulting increase in temperature will be

$$t = \frac{14{,}544}{3\frac{2}{3} \times 0.217} = 18{,}282 \text{ degrees F.}$$

Such a temperature, of course, cannot be reached in practice, even with pure carbon and oxygen, on account of the radiation losses.

In the ordinary furnace the oxygen is obtained from the atmosphere, which, neglecting a few impurities and minor elements, contains the following, mechanically mixed:

Oxygen	23 parts by weight.
Nitrogen	77 parts by weight.
or	
Oxygen	21 parts by volume.
Nitrogen	79 parts by volume.

Hence in the combustion of one pound of pure carbon the products of combustion contain not only $3\frac{2}{3}$ pounds of CO_2, but $\frac{77}{23} \times 2\frac{2}{3} = 8.92$ pounds of nitrogen, giving a total of $3\frac{2}{3} + 8.92 = 12.58$ pounds.

The nitrogen performs no useful office in combustion and passes through the furnace without change. It dilutes the products of combustion and reduces their temperature. Thus in the specific case taken above the theoretical increase in temperature will be

$$t = \frac{14,544}{12.58 \times 0.236} = 4900 \text{ degrees F.}$$

The mean specific heat, 0.236, of the products of combustion is determined as follows:

Heat necessary to raise 8.92 lbs. of nitrogen 1 degree is 8.92×0.2438 (specific heat of nitrogen) = 2.1747 B.T.U.

Heat necessary to raise 3.66 lbs. of CO_2 1 degree is 3.66×0.217 (specific heat of CO_2) = 0.7942 B.T.U.

Total heat = 2.1747 + 0.7942 = 2.9689 B.T.U.

$$\text{Mean specific heat} = \frac{2.9689}{8.92 + 3.66} = 0.236.$$

Evidently for maximum temperature this dilution should be kept as low as possible. Unfortunately, in practice, a perfect union of fuel and air in theoretical proportions is almost impossible, and to insure complete combustion an excess of air is necessary. This still further reduces the temperature of the products of combustion. For example, the complete oxidation of one pound of carbon requires 11.5 pounds of air. If 20 pounds of air are supplied per pound of carbon, an average figure in good practice, the theoretical increase in temperature will be

$$t = \frac{14,544}{21 \times 0.236} = 2934 \text{ degrees F.}$$

In the preceding computation the specific heat of the constituent gases is assumed to be constant for all temperatures. Recent experiment seems to show that the specific heat increases at high temperatures, though actual values are not yet available.

20. Air required for Combustion. — The quantity of air required for complete combustion may be approximately determined from the ultimate analysis. Thus

$$A = 34.56\left(\frac{C}{3} + H - \frac{O}{8}\right), \tag{2}$$

in which

A = weight of air required per pound of fuel.

C, H, and O = proportional part by weight of carbon, hydrogen, and oxygen in the fuel.

When the flue-gas analysis is known, the air actually supplied may be approximately determined as in the following example:

Example: Required the weight of air supplied per pound of coal with fuel and flue gas analysis as follows:

Coal analysis (Illinois bituminous, mine run):

Carbon	68.14	Nitrogen	1.34
Oxygen	15.83	Ash	7.66
Hydrogen	5.38	Sulphur	1.65

Flue gas analysis, cubic feet per 100 cubic feet of dry gas (temperature, 62° F., barometer 30 inches):

CO_2	14
CO	0.5
O	6.0
SO_2	(not determined, since its weight is practically negligible).

The weights may be determined from the densities given in Table 2.

	Volume.	Density.	Weight.
CO_2	14	0.1159	1.6226
O	6	0.0843	0.5058
CO	0.5	0.0737	0.0368

These weights can be subdivided into those of their constituents; thus the CO_2 contains $\frac{3}{11}$ of carbon and $\frac{8}{11}$ of oxygen, and the CO, $\frac{3}{7}$ of carbon and $\frac{4}{7}$ of oxygen.

$$\begin{aligned}\tfrac{8}{11} \times 1.6226 &= 1.1800\\ \tfrac{4}{7} \times 0.0368 &= 0.0210\\ &\ 0.5058\\ \text{Pounds of oxygen}\ &\ 1.7068\end{aligned}
\qquad
\begin{aligned}\tfrac{3}{11} \times 1.6226 &= 0.4426\\ \tfrac{3}{7} \times 0.0368 &= 0.0158\\ \text{Pounds of carbon}\ &\ 0.4584\end{aligned}$$

$$\text{Weight of oxygen per pound of carbon} = \frac{1.7068}{0.4584} = 3.72.$$

Since 23% of air by weight is oxygen,

$$\text{Weight of air per pound of carbon} = \frac{3.72}{0.23} = 16.2.$$

Weight of air supplied per pound of coal is

$$\begin{aligned}&\text{for the carbon,} && 0.6814 \times 16.2 = 11.04 \text{ lbs.}\\ &\text{for the hydrogen,} && 34.56\left(0.0538 - \frac{0.1583}{8}\right) = 1.17 \text{ lbs.}\\ &\text{total} && 12.21 \text{ lbs.}\end{aligned}$$

The theoretical weight of air for combustion would have been

$$11.58 \times 0.6814 + 34.56\left(0.0538 - \frac{0.1583}{8}\right) = 9.06 \text{ lbs.}$$

$$\text{The percentage of air excess} = 100\,\frac{12.21 - 9.06}{9.06} = 35.$$

The weight of air actually supplied may be closely approximated by the following equation:

$$A = 3.032\left(\frac{N}{CO_2 + CO}\right) \times C, \tag{2a}$$

in which A = the weight of air supplied per pound of fuel.

N, CO, CO_2 = percentages by volume of nitrogen, carbon monoxide, and carbon dioxide in the flue gas.

C = the proportional part by weight of carbon in the fuel.

The per cent of excess air supplied per pound of *fuel* may be conveniently determined from the relationship

$$\frac{\text{Air actually required}}{\text{Air theoretically supplied}} = \frac{N}{N - 3.782\,O}.$$

N and O are respectively, by volume, the proportional parts of the nitrogen and oxygen in the flue gas. The free oxygen is due to the air supplied and not used. This oxygen was accompanied by 3.782 times its volume of nitrogen. (N − 3.782) represents the nitrogen content in the air actually required for combustion. Hence, N ÷ (N − 3.782) is the ratio of the air supplied to that required.

Table 2a gives the values of the ratio corresponding to various percentages of CO_2 + CO and CO_2 + CO + O.

TABLE 2a.

RATIO OF TOTAL AIR SUPPLIED TO THAT THEORETICALLY REQUIRED FOR VARIOUS ANALYSES OF FLUE-GASES.

$$\text{Ratio} = \frac{N}{N - 3.782\,O}.$$

CO_2+CO.	N=79. CO_2+CO +O=21.	N=79.5. CO_2+CO +O=20.5.	N=80. CO_2+CO +O=20.	N=80.5. CO_2+CO +O=19.5.	N=81. CO_2+CO +O=19.	N=81.5. CO_2+CO +O=18.5.	N=82. CO_2+CO +O=18.
21	1.02	....	....	....	....	....	....
20	1.05	1.00	1.00	....	....	....	....
19	1.11	1.08	1.05	1.02	1.00	....	....
18	1.17	1.14	1.10	1.08	1.05	1.02	1.00
17	1.24	1.20	1.17	1.13	1.10	1.07	1.05
16	1.32	1.27	1.23	1.20	1.16	1.13	1.10
15	1.40	1.35	1.31	1.27	1.23	1.19	1.16
14	1.51	1.45	1.39	1.35	1.30	1.26	1.23
13	1.62	1.55	1.50	1.44	1.39	1.34	1.30
12	1.76	1.68	1.61	1.54	1.49	1.43	1.38
11	1.92	1.82	1.74	1.66	1.60	1.53	1.48
10	2.11	2.00	1.90	1.81	1.72	1.65	1.59
9	2.35	2.21	2.08	1.97	1.88	1.79	1.71
8	2.65	2.47	2.31	2.18	2.06	1.95	1.86
7	3.03	2.80	2.59	2.44	2.27	2.14	2.03
6	3.55	3.22	2.96	2.74	2.54	2.38	2.24
5	4.27	3.81	3.44	3.14	2.89	2.68	2.50
4	5.37	4.65	4.11	3.68	3.34	3.05	2.83
3	7.23	5.97	5.10	4.45	3.96	3.56	3.25
2	11.06	8.34	6.71	5.63	4.85	4.27	3.82
1	23.51	13.83	9.83	7.64	6.27	6.12	4.64

The weight of air per pound of combustible as indicated by the per cent of CO_2 in the flue gas is given in Table 3. These figures are only approximate but are sufficiently accurate for many practical purposes.

TABLE 3.

WEIGHT OF AIR PER POUND OF COMBUSTIBLE AS INDICATED BY THE PERCENTAGE OF CO_2 IN THE FLUE GAS.

Per Cent of CO_2.	Pounds of Air.	Per Cent of CO_2.	Pounds of Air.	Per Cent of CO_2.	Pounds of Air.
21	12	14	18	7	36
20	12.6	13	19.4	6	42
19	13.3	12	21	5	50.5
18	14	11	22.9	4	63
17	14.8	10	25.2	3	84
16	15.7	9	28	2	126
15	16.8	8	31.5	1	210

21. Calorific Value of Coal. — The heat liberated by the combustion of unit weight of fuel is called the calorific value of the fuel. The most rational way of determining the heat of combustion is to burn a weighed sample of coal in an atmosphere of oxygen in a suitable calorimeter. An alternative method is to calculate the heat of combustion from the chemical analysis. An analysis which determines the per cent of fixed carbon, volatile matter, moisture, and ash is called the *proximate analysis*, while one which reduces the fuel to its elementary constituents of carbon, hydrogen, nitrogen, sulphur, moisture, and ash is called the *ultimate analysis*. The proximate analysis is comparatively easy to make and gives the general characteristics of a fuel. It is made by subjecting the sample to a moderate temperature to expel the moisture, then to a higher temperature until the volatile matter is driven off, and finally to a very high temperature which drives off all carbon as carbon dioxide and leaves the ash as a residue. By weighing the residue at the end of each operation the various percentages may be computed. For method of making proximate analysis, see "Report of Committee on Coal Testing," *Journal of the American Chemical Society*, Vol. 21, p. 1116.

The heating value of certain classes of coals may be estimated from the proximate analysis. Thus for Illinois coals with ash content under 10 per cent, R. W. Hunt & Co. deduced the formula

$$h = 14{,}544\,C + 16{,}515\,V - 10{,}000\,A, \qquad (3)$$

in which

h = B.T.U. per pound of coal.

C, V, and A = the proportional content of fixed carbon, volatile matter, and ash.

When ash lies between 10 and 15 per cent, the formula will be more accurate if written

$$h = 14{,}544\ C + 16{,}515\ V + 354\ A - 1635 \qquad (4)$$

Kent, Goutal, and other authorities have deduced formulas based on the proximate analysis which agree closely with the calorimetric determinations for many coals, but a calorimetric determination is necessary whenever exact results are required.

The formula most commonly used in calculating the heating value of a fuel is based on the ultimate analysis and is known as Dulong's formula. Thus

$$h = 14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}000\ S, \qquad (5)$$

in which

h = heating value in B.T.U. per pound of coal.

C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur, respectively.

The calorimetric determination is more readily made and is usually more reliable than the calculation from the ultimate analysis. Table 1 gives the proximate and ultimate analyses and the calorimetric and calculated heat values for a number of American coals.

Calorific Value of Fuels: Poole, Engr. U.S., April 1, 1903; P. Mahler, Jour. Frank. Inst., Jan., 1905; A. Adams, Jour. Soc. Chem. Ind., Oct. 31 and Nov. 30, 1901; Power, Oct. 1, 1906; Mines and Min., May, 1902; Trans. Am. Inst. Min. Engrs., 27–259; Kent, Steam Boiler Economy, Chap. V; Report of Government Coal Testing Plant at St. Louis.

Calorimetry: Berthier Method of Coal Calorimetry: Trans. A.S.M.E., 21–304. *Comparison of Calorimeters:* Jour. Soc. Chem. Ind., 22–1230, 1903; 23–704. *Improved Form of Thompson Calorimeter:* Jour. Soc. Chem. Ind., 25–409, 1906. *Parr's Calorimeter:* Chem. Engr., Feb., 1907. *Coal Calorimeters:* Trans. A.S.M.E., 14–816, 16–1040. *Mahler Bomb: Calorific Power of Fuels,* Poole; Trans. Am. Inst. Min. Engrs., 27–259, Engr. U.S., Jan. 1, 1907, p. 68.

22. Heat Losses in Burning Coal. — The function of the boiler is the absorption of the heat liberated by the combustion of the fuel. In practice from 50 per cent to 85 per cent of this heat is utilized in making steam, depending upon the conditions of operation, and the remainder is wasted. Complete utilization of the heat of combustion is impossible. A boiler and furnace which utilizes 80 per cent of the heat in the fuel is exceptional and an average figure for very good practice is not

far from 75 per cent. The various losses may be summed up as follows:

1. Loss in the dry chimney gases.
2. Loss due to incomplete combustion.
3. Loss of fuel through the grate.
4. Superheating the hygroscopic moisture in the air.
5. Moisture in the fuel.
6. Loss due to the presence of hydrogen in the fuel.
7. Unburned fuel carried beyond the combustion chamber in the form of soot or smoke.
8. Radiation and minor losses.

Fuel Losses: G. H. Barrus, Cassier's Mag., Aug., 1907; Power, Sept. 14, 1909, p. 439.

23. Loss in the Dry Chimney Gases. — The most serious loss is the heat carried away in the chimney gases. This may be expressed

$$h = (W + 1)(T - t)S, \tag{6}$$

in which

h = loss in B.T.U. per pound of combustible.
W = weight of air supplied per pound of combustible.
T = chimney temperature, degrees F.
t = temperature of the air entering furnace.
S = specific heat of the gases. (This may be taken as 0.24 for most purposes.)

This loss is unavoidable and is seldom less than 15 per cent of the total heat supplied. In average good practice it is not far from 20 per cent. Even in the ideal case when pure carbon burns to CO_2 and the air supply is theoretically correct, the loss is appreciable.

For example, suppose the temperature of the air is 50 degrees F., the temperature of the flue gas 450 degrees F., and the air supply 11.5 pounds per pound of combustible, the heat carried away by the chimney will be

$$12.5 \times (450 - 50) \times 0.24 = 1200 \text{ B.T.U.},$$

which is $1200 \div 14,500 = 8.3$ per cent of the total heat supplied.

As a matter of fact, considerable excess of air is necessary for complete combustion, depending upon the size and nature of the fuel, thickness of the fire, variation in resistance through the fuel, intensity of draft, and character of grate. Practice shows a minimum excess of 50 per cent in well-proportioned furnaces working under ideal conditions with an average not far from 100 per cent.

Table 4 indicates the magnitude of these losses for different chimney temperatures and weights of air per pound of combustible.

TABLE 4.

HEAT CARRIED AWAY BY THE DRY CHIMNEY GASES PER POUND OF COMBUSTIBLE.

Pounds of Air per Pound of Combustible.	Temperature of Chimney Gases. Deg. Fahr.							
	300°	350°	400°	450°	500°	550°	600°	650°
12 *	750 5.2	905 6.2	1060 7.3	1216 8.7	1370 9.5	1528 10.5	1683 11.6	1840 12.7
15	865 6	1112 7.6	1305 9.1	1498 10.3	1679 11.6	1880 13.0	2072 14.3	2262 15.6
18	1004 7.2	1321 9.1	1550 10.7	1778 12.2	2010 13.9	2235 15.4	2460 17	2692 17.9
21	1266 8.7	1530 10.5	1785 12.3	2060 14.2	2320 16	2582 17.8	2846 19.5	3118 21
24	1440 9.9	1740 12	2040 14	2340 16.1	2640 18.2	2940 20.3	3240 22.4	3540 24.4
27	1611 11.1	1950 13.5	2281 15.7	2620 18.1	2958 20.4	3291 22.7	3628 25	3962 27.4
30	1785 12.4	2160 14.9	2530 17.4	2900 20	3270 22.6	3641 25	4016 27.8	4396 30.4
33	1957 13.5	2362 16.3	2779 19.2	3180 22	3589 24.7	4000 27.6	4405 30.5	4820 33.2
36	2130 14.7	2579 17.8	3020 20.8	3461 23.9	3910 27	4350 30	4798 33	5290 36.6
39	2300 15.9	2781 19.2	3261 22.5	3743 25.8	4220 29.2	4700 32.4	5180 35.7	5670 39
42	2479 17.1	2999 20.6	3508 24.7	4023 27.7	4540 31.3	5052 34.8	5570 39.4	6100 42

* Theoretical requirement.

Large type gives the loss in B.T.U. per pound of combustible.

Small type gives the per cent loss, assuming a calorific value of 14,500 B.T.U. per pound of combustible.

From the table it will be seen that this loss is approximately 12 per cent of the total heat supplied when the excess of air is 50 per cent and 16 per cent when the excess is 100 per cent. Flue-gas temperature assumed to be 450 degrees F. Flue temperatures less than 450 degrees F. are seldom experienced except in connection with economizers, and an average figure is about 500 degrees F.

The weight of air supplied per pound of combustible is most conveniently determined by the percentage of CO_2 in the flue gas. Thus for the complete oxidation of pure carbon the resulting flue gases

TABLE 5.

LOSS DUE TO INCOMPLETE COMBUSTION OF CARBON TO CARBON MONOXIDE.

Per Cent of CO in the Flue Gas by Volume.	Per Cent of CO_2 in the Flue Gas by Volume.					
	6	8	10	12	14	16
0.2	**328** 2.2	**248** 1.7	**199** 1.3	**168** 1.1	**144** 1	**126** 0.8
0.4	**635** 4.3	**484** 3.3	**390** 2.6	**327** 2.2	**282** 1.9	**248** 1.7
0.6	**925** 6.3	**709** 4.8	**575** 3.9	**474** 3.2	**417** 2.8	**367** 2.5
0.8	**1192** 8.1	**923** 6.3	**750** 5.1	**635** 4.3	**549** 3.7	**495** 3.4
1.0	**1494** 10.2	**1128** 7.7	**923** 6.3	**780** 5.3	**676** 4.6	**596** 4.1
1.2	**1690** 11.5	**1321** 9	**1085** 7.4	**923** 6.3	**801** 5.4	**708** 4.8
1.4	**1920** 13.1	**1512** 10.3	**1248** 8.5	**1061** 7.2	**924** 6.3	**819** 5.6
1.6	**2104** 14.3	**1693** 11.5	**1400** 9.5	**1193** 8.1	**1040** 7.1	**924** 6.3
1.8	**2340** 16	**1865** 12.7	**1549** 10.5	**1321** 9.0	**1151** 7.8	**1025** 7
2.0	**2537** 17.2	**2030** 13.8	**1690** 11.5	**1450** 9.9	**1270** 8.6	**1129** 7.7

Large type gives the loss in B.T.U. per pound of carbon. Small type gives the per cent loss, assuming a calorific value of 14,650 B.T.U. per pound of carbon.

should consist of carbon dioxide and nitrogen only, and in the ratio by volume of 21 to 79; therefore 21 per cent of CO_2 in the flue gas is indicative of complete combustion and theoretical air supply. In other words, the ratio by volume of CO_2 to N after complete combustion is practically the same as the ratio of the oxygen to the nitrogen in the air before combustion. Table 3 gives the approximate weight

of air used per pound of combustible for different percentages of CO_2 in the flue gas.

In practice, 15 per cent is all that can be expected under the *best* conditions, with an *average* between 10 per cent and 12 per cent. Anything less than 10 per cent shows an excessive amount of air supplied. Traveling grates, unless carefully operated, are apt to show as low as 5 per cent of CO_2.

24. Loss due to Incomplete Combustion. — If the volatile gases are not completely oxidized, as when the air supply is insufficient or the mixture of air and gases is not thorough, some of the carbon may escape as CO. The presence of even a small amount of CO in the flue gas is indicative of a very appreciable loss, as will be seen from Table 5. Carbon monoxide is a colorless gas, and its presence in the chimney gases cannot be detected by the fireman, consequently the absence of smoke is not an infallible guide for perfect combustion. This loss may be expressed

$$h_2 = C \times \frac{10{,}150\ CO}{(CO_2 + CO)}, \tag{7}$$

in which $h_2 =$ the loss in B.T.U. per pound of carbon, CO_2 and CO are percentages by volume of the flue gases and C is the proportional part of carbon in the combustible.

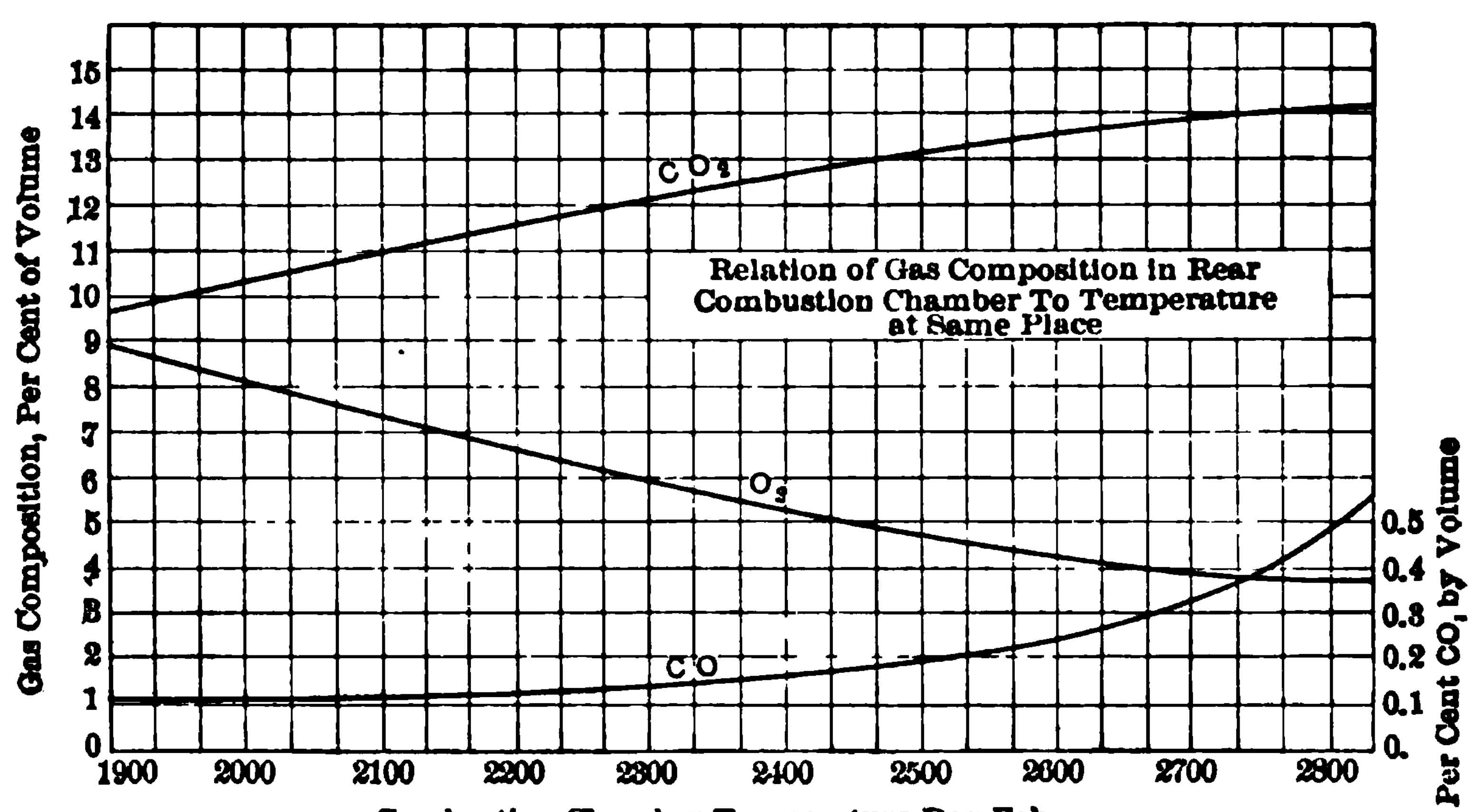


FIG. 5. Relation of Gas Composition in Combustion Chamber to Temperature.

This loss, however, may be wholly avoided in a properly designed and carefully operated furnace. In fact the loss from this cause is often exaggerated and seldom exceeds 2 per cent of the total heat value of the fuel except during the few moments following the replenishing

of a burned-down fire with fresh fuel or when the supply of air is checked to meet a sudden reduction in load. In improperly designed furnaces in which the volatile gases are brought into contact with the cooler boiler surface before combustion is complete, the carbon monoxide may be reduced in temperature below its ignition point and consequently will fail to combine with the oxygen. In such a case the loss may prove to be a serious one. Fig. 5 shows the relation between the composition of the products of combustion in the rear combustion chamber of a 250-horse-power Heine boiler, hand fired, and the temperature at the same place. (For an extended discussion of this subject see Jour. West. Soc. Engrs., June, 1907, p. 285.)

25. Loss of Fuel through Grate. — The refuse from a fuel is that portion which falls into the pit in the form of ashes, unburned or partially burned fuel, and cinders. The loss from this cause depends upon the size of the fuel, the width of opening in the grate bars, and the type of grate. Coal which necessitates frequent slicing is apt to give greater loss than a free-burning coal. Under good conditions of operation it ought not to exceed 2 per cent of the total heat value of the fuel. In traveling grates in which a large percentage of the fine fuel falls through the front end of the grate a special hopper is ordinarily installed in the ash pit which reclaims most of it. (See Fig. 99.)

Loss of Fuel in Ashes: Power, March, 1905. *Experiments on Fuel Value of Bituminous Coal Ashes:* Technology Quarterly, Dec., 1905. *Coal Ash:* Jour. Soc. Chem. Ind., Jan. 15, 1904.

26. Superheating the Moisture in the Air. — The loss due to this cause is a minor one, though on hot, humid days it may be appreciable. This loss may be expressed

$$h_3 = 0.48\, M\, (T - t),^* \qquad (8)$$

in which h_3 = B.T.U. lost per pound of combustible.

M = weight of moisture introduced with the air per pound of combustible.

t = temperature of air entering the furnace, degrees F.

T = temperature of chimney gases, degrees F.

27. Moisture in the Fuel. — Moisture in the fuel represents an appreciable loss in economy if present in large quantities, since the heat necessary to evaporate it into superheated steam at chimney temperature is lost. Firemen occasionally wet the coal to assist coking or to reduce

* The latest accepted value for the mean specific heat of water vapor at atmospheric pressure is 0.46 in place of 0.48.

the dust, but moisture thus added necessarily reduces the furnace efficiency. The loss due to this cause is expressed:

$$h_4 = M\,[(212 - t) + 966^* + 0.48^\dagger\,(T - 212)], \qquad (9)$$

in which h_4 = B.T.U. lost per pound of combustible.
M = weight of moisture per pound of combustible.

Other notations as in preceding equation.

For example, the heat loss due to the moisture in a pound of fuel containing 10 per cent water, temperature of fuel 80 degrees F., chimney temperature 480 degrees F., is

$$h_4 = 0.1\,[(212 - 80) + 966^* + 0.48^\dagger\,(480 - 212)]$$
$$= 122.6 \text{ B.T.U.}$$

A rough rule is to allow a loss of 1 per cent of the total heat value of the dry fuel for each 10 per cent of moisture present.

28. Loss due to the Presence of Hydrogen in the Fuel. — The hydrogen in any fuel which is not rendered inert by oxygen burns to water and in so doing liberates 62,032 B.T.U. per pound. All of this heat is not available for producing steam in the boiler, since the water formed by combustion is discharged with the flue gases as superheated steam at chimney temperature. This loss is equal to

$$h_5 = 9\,H\,[(212 - t) + 966^* + 0.48^\dagger\,(T - 212)], \qquad (10)$$

in which h_5 = B.T.U. lost per pound of combustible.
H = weight of hydrogen per pound of combustible.

All other notations as in equations (8) and (9).

With anthracite coal this loss is approximately 2.5 per cent of the total heat value of the combustible and with bituminous coal it runs as high as 4.5 per cent.

29. Loss due to Smoke. — Visible smoke consists of carbon in a flocculent state mixed with the products of combustion. It is seldom evident in connection with anthracite coal and is generally associated with bituminous fuel. A smoky chimney does not necessarily indicate an inefficient furnace, since the losses due to visible smoke generation seldom exceed 2 per cent; as a matter of fact, a smoky chimney may be much more economical than one which is smokeless. That is to say, a furnace operating with minimum air supply may cause dense clouds of smoke and still give a higher evaporation than one made smokeless by a very large excess of air. There will be some loss due to carbon monoxide and unburned carbon or soot in the former case, but this may be more than offset by the excessive losses caused by the

* See footnote, p. 88. † See footnote, p. 37.

heat carried away in the chimney gases in the latter. Judging from the results of the majority of steam power plants using bituminous coal, even those recently installed, smokeless and efficient combustion is not readily effected and the problem is far from being satisfactorily solved.

Smoke has become such a public nuisance, particularly in the larger cities, that special ordinances prohibiting its production have been enacted and violators are subject to heavy fines. Effective enforcement of these ordinances renders smoke production very costly and the problem of smokeless combustion becomes a momentous one.

The subject of smoke prevention and smoke-prevention devices is discussed at some length in Chapter V.

30. Radiation and Minor Losses. — These losses are usually determined by difference. That is, the difference between the heat represented in the steam and the losses just mentioned is charged to radiation, leakage, and unaccounted for. Summing up the various losses we have

	Excellent Practice. Per Cent.	Good Practice. Per Cent.	Average Practice. Per Cent.	Poor Practice. Per Cent.
Heat given to steam	80	70	60	50
Loss in chimney gases	12	18	24	30
Loss due to carbon burning to CO	0	1	2	3
Loss of fuel through grate	0.5	1	2	3
Loss due to moisture in coal, moisture in air, and hydrogen in fuel	3.0	3	3	3.5
Smoke, soot, etc.	0	0.5	1	1.5
Radiation and minor losses	4.5	6.5	8	9

31. Size of Coal. — Bituminous. Coal is usually marketed in different sizes, ranging from *lump coal* to *screenings*. The latter furnish by far the greater part of the stoker fuel used. For maximum efficiency coal should be uniform in size. With hand-fired furnaces there is usually no limit to its fineness and larger sizes can be used than with stokers. As a rule the percentage of ash increases as the size of coal decreases. This is due to the fact that all of the fine foreign matter separated from larger coal, or which comes from roof or floor of the mine, naturally finds its way into the smaller coal. The size best adapted for a given case is dependent upon the intensity of draft, kind of stoker or grate, and the method of firing, and its proper selection often affords an opportunity to effect considerable economy. Some idea of the influence of the size of screenings on the capacity and efficiency of a

boiler in a specific case is illustrated in Fig. 6. The curves are plotted from a series of tests conducted with Illinois screenings on a 500-horse-power B. & W. boiler, equipped with chain grates, at the power house of the Chicago Edison Company.

Influence of Thickness of Fire. — See paragraph 76.

Size of Coal: Some Characteristics of Coal as affecting Performances with Steam Boilers: Jour. West. Soc. Engrs., Oct., 1906, p. 528. *Small Size Anthracite:* Eng. and Min. Jour., Dec. 22, 1904. *The Economy of Small Size Coals for Power Plants:* Eng. Mag., Feb., 1905.

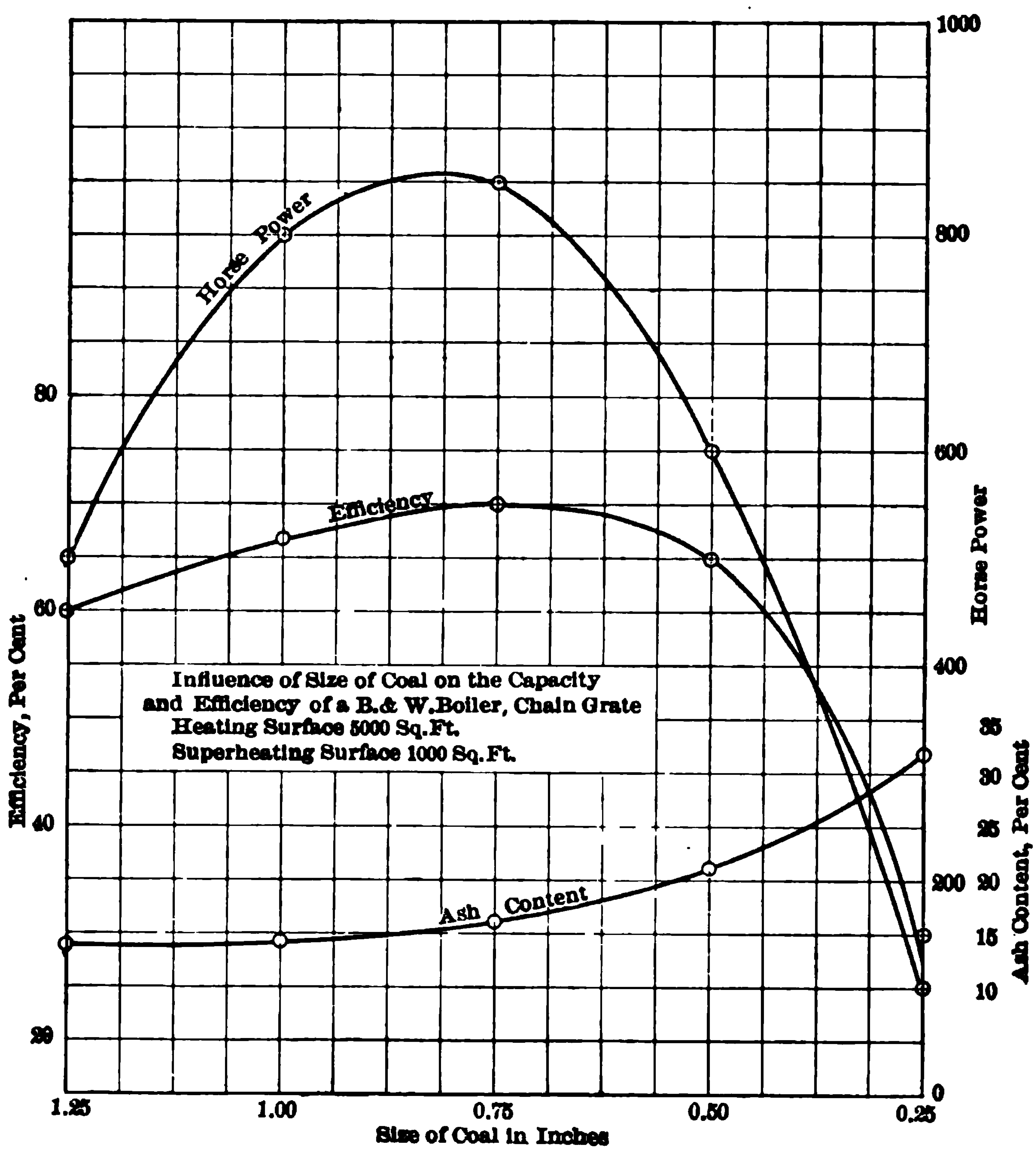


FIG. 6. Influence of Size of Coal on Boiler Capacity and Efficiency.

32. Washed Coal. — The washing of coal is for the purpose of separating from it such impurities as slate, sulphur, bone coal, and ash. All of these impurities show themselves in the ash when the coal is burned. Screenings contain anywhere from 5 per cent to 25 per cent

of ash and from 1 per cent to 4 per cent of sulphur. Washing eliminates about 50 per cent of the ash and some of the sulphur. Table 6 gives some idea of the effects of washing upon a number of grades of coal. The evaporative power of the combustible is practically unaffected by washing and the greater part of the water taken up by the coal is removed by thorough drainage. Many coals otherwise worthless as steam coals are rendered marketable by washing. Washed coals are usually graded as follows:

Size.	Screens.	
No. 1	Over $1\frac{3}{4}$	Under $2\frac{1}{2}$
2	$1\frac{1}{8}$	$1\frac{3}{4}$
3	$\frac{3}{4}$	$1\frac{1}{8}$
4	$\frac{1}{4}$	$\frac{3}{4}$
5	...	$\frac{1}{4}$

Numbers 3 and 4 are excellent sizes for use in connection with stokers and No. 5 is well adapted for hand furnaces where smoke prevention is essential.

TABLE 6.

EFFECT OF WASHING ON BITUMINOUS COALS.

(*Journal W.S.E.*, December, 1901.)

	Before Washing. (Per Cent.)			After Washing. (Per Cent.)		
	Ash.	Sulphur.	Fixed Carbon.	Ash.	Sulphur.	Fixed Carbon.
Belt Mountain, Mont.	18.74	3.34	43.72	5.56	2.40	48.39
Wellington Colliery Co., Vancouver Island (new coal)	35.00	......	38.00	8.90	......	56.90
Alexandria Coal Co., Crabtree, Pa.	10.60	1.30	....	6.21	0.61	......
DeSoto, Ill.	18.00	......	44.00	4.20	......	57.00
Northwestern Improvement Co., Roslyn, Wash.	16.30	0.57	45.90	9.70	0.40	47.86
Luhrig Coal Co., Zaleski, Ohio	15.80	1.90	....	8.00	0.87	50.90
Rocky Ford Coal Co., Red Lodge, Mont.	25.30	......	37.80	8.50	......	47.20
Buckeye Coal and Ry. Co., Nelsonville, Ohio	13.77	1.05	49.04	4.30	0.89	54.82
New Ohio Washed Coal Co., Carterville, Ill.	9.48	0.78	55.00	4.85	0.69	63.00

Modern Method of Coal Washing: Eng. and Min. Jour., May 9, 1903, Oct. 13, 1904. *Principles of Coal Washing:* Mines and Min., Aug., 1903. *Bituminous Coal Washing:* Mines and Min., April, 1905. *Washing of Bituminous Coals by Luhrig Process:* Jour. West. Soc. Engrs., Dec., 1901. *Coal Washing:* Jour. Soc. Chem. Ind., April 30, 1904. *Anthracite Washeries:* Eng. and Min. Jour., April 28, 1906; Col. Guard, April 20, 1906; Trans. Am. Inst. Min. Engrs., Nov., 1905. *Studies on Coal Washing:* Col. Guard, Nov. 21, 1902. *Coal Washing by Stuart System:* Mines and Min., Dec., 1903. *Coal Washing at Collinsville, Ill.:* Mines and Min., Sept., 1901. *Bellevue Washery of D. L. and W. R.R. Co., Scranton, Pa.:* Mines and Min., June, 1903. *Coal Washery at Howe, Indian Territory:* Mines and Min., March 14, 1904. *Eastern Coal and Coke Co.'s Washery at Kansas:* Eng. and Min. Jour., Sept. 20, 1902.

33. Purchasing Coal.*— Engineers fail to agree as to the specifications best suited for the purchase of coal. Some extensive purchasers require elaborate analyses and others specify only the size and grade of the fuel. Every essential requirement of the purchaser may be fulfilled by confining them to the four following characteristics:

Moisture.
Ash.
Size of coal.
Calorific value of the coal.

Although moisture is a great and uncertain variable, and the producer can exercise no control over this factor, still the purchaser should protect himself against excessive moisture by stipulating an amount consistent with the average inherent moisture in the coal, and proper penalty should be fixed for delivery in excess of the amount allowed, a corresponding bonus being paid for delivery of less than contract amount. Considerable attention should be given to the percentage of earthy matter contained. The amount of earthy matter usually fixes the heating value of the coal, since the heating value of the combustible is practically constant. The effect of ash on the heat value of Illinois screenings as fired under a B. & W. boiler with chain grate is shown in Fig. 7. This value varies with the different types of boilers, grates, and furnaces, but is substantially as illustrated. The amount of refuse in the ash pit is always in excess of the earthy matter as reported by analysis.

The maximum allowable amount of sulphur is sometimes specified, since some grades of coal high in sulphur cause considerable clinkering. But sulphur is not always an indication of a clinker-producing ash, and a more rational procedure would be to classify a coal as clinkering or non-clinkering according to its behavior in the particular furnace in question, irrespective of the amount of sulphur present. An analysis of the various constituents of the ash is necessary to

* See also Selection of Coal for Boiler Furnaces, by D. T. Randall, Power & Engr., Apr. 6, 1909, p. 642.

determine whether or not the sulphur unites with them to produce a fusible slag, and as such analyses are usually out of the question on account of the expense attached, they may well be omitted.

The heating value of the coal as determined by a sample burned in an atmosphere of oxygen does not give its evaporative power, since this

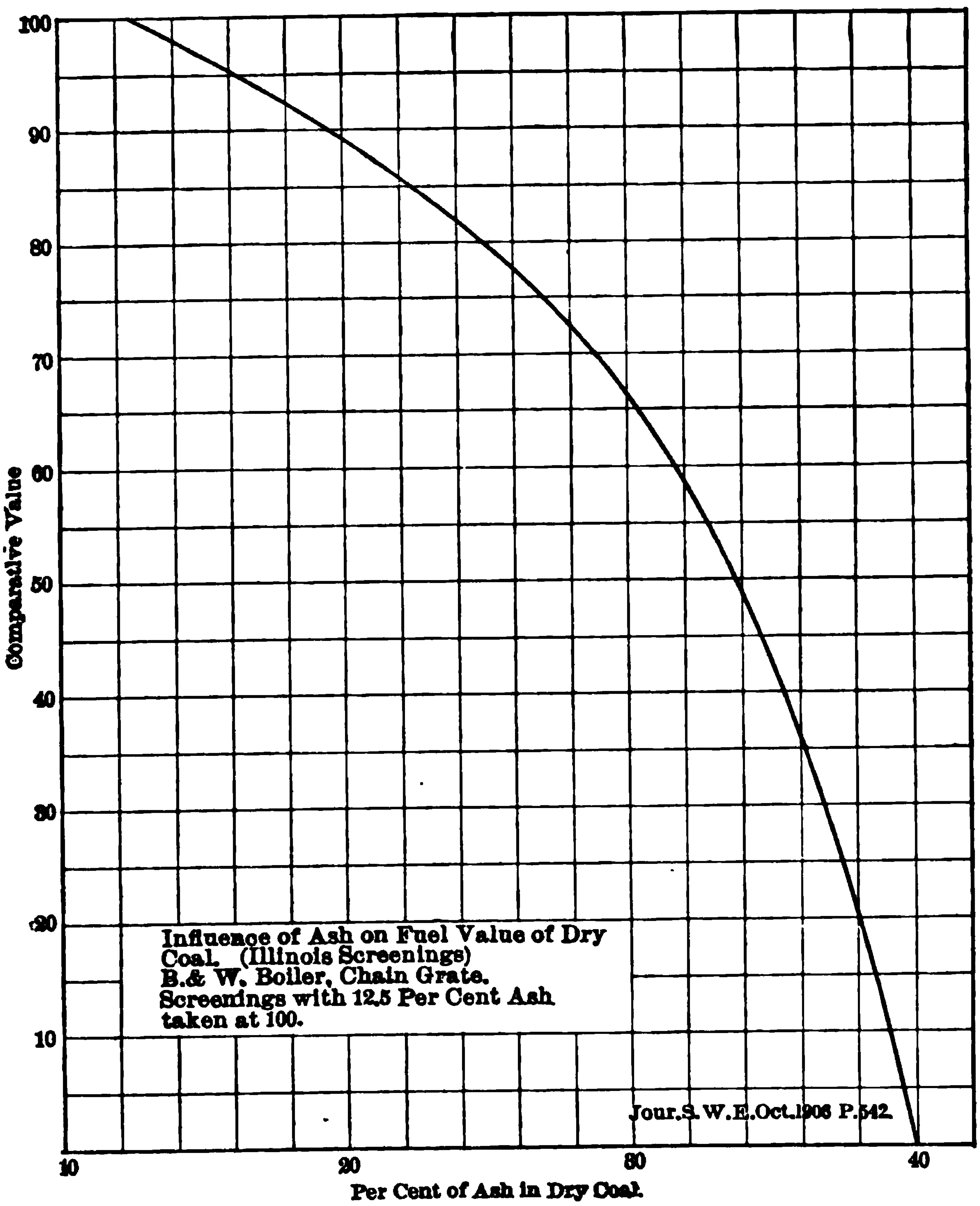


FIG. 7. Influence of Ash on Fuel Value of Dry Coal.

depends largely upon the composition of the fuel, character of grate, and conditions of operation. It merely serves as a basis upon which to determine the efficiency of the furnace. In large plants where a number of grades of fuel are available it is customary to conduct a series of tests with the different grades and sizes, and the one which evaporates

the most water for a given sum of money, other conditions permitting, is the one usually contracted for. In designing a new plant particular attention should be paid to the performance of similar plants already in operation, and that fuel and stoker should be selected which are found to give the best returns for the money. Where smoke prevention is a necessity the smoke factor greatly influences the choice of fuel and stoker.

See paragraph 416.

Testing and Purchasing Coal for Steam Plants: Eng. News, Feb. 7, 1907; Eng. Rec., Sept. 22, 1906, p. 326; Engr. U.S., Aug. 15, 1907; Bulletin No. 339 U.S. Geological Survey, 1908. *Coal for Hand-Fired Furnaces:* Nat. Engr., July, 1909.

34. Powdered Coal. — The value of powdered coal as a fuel for steam boiler plants has long been known, and appliances for pulverizing and feeding the coal have been on the market for a number of years. However, despite the many advantages of powdered fuel and the apparent success of some of the systems of burning it, little progress has been made toward its general adoption.

Some of the advantages obtained in burning powdered coal are:

a. Complete combustion and total absence of smoke. The coal in the form of dry impalpable dust is induced or forced into the zone of combustion, where each minute particle is brought into contact with the necessary amount of air and complete oxidation is effected without the excess of air which accompanies the firing with lump coal, provided the furnace is properly proportioned. With a properly designed setting there is complete absence of smoke.

b. A cheaper grade of bituminous coal may be burned, since the per cent of ash and moisture has little effect on the completeness of combustion and the full value of the fuel is more nearly realized than with ordinary firing.

c. The plant may be rapidly forced above its rated capacity and sudden demands for power readily met.

d. The labor of firing is reduced to a minimum.

Pulverized Fuel: Eng. Mag., Jan., 1908; Jour. West. Soc. Engrs., Feb., 1904; Am. Elecn., Sept., 1901. *Coal Dust for Steam Making:* Engr. U.S., Feb. 15, 1899. *Burning Pulverized Coal:* Eng. and Min. Jour., Dec. 31, 1903, May 12, 1906; Jour. Assn. Eng. Soc., July, 1903. *Use of Pulverized Coal under Steam Boilers:* Eng. News, April 1, 1904; Power, March, 1904, April, 1904. *Coal Dust Firing:* Eng. and Min. Jour., Dec. 16, 1905. *Coal Dust Fuel:* Engr., Lond., Jan. 31, 1896; Engr. U.S., April 1, 1903; Eng. News, Feb. 20, 1902.

35. Depreciation of Powdered-Coal Furnaces. — To withstand the intense heat of combustion, brickwork of the highest quality is essential, since common fire brick are soon reduced to a liquid slag. A good quality

of fire brick will withstand the heat for several months without renewals provided the furnace is properly enclosed, otherwise the strain of expansion and contraction due to alternate heating and cooling will crack the brick. Excellent results have been obtained from the use of bricks composed chiefly of the refuse from a carborundum slag, but the high cost has prevented their general use.

36. Storing Powdered Fuel. — Most cities limit the storage of powdered coal to such a small quantity as to prohibit the use of fuel feeders of the "dust feed" type in plants of any size not provided with a pulverizing and crushing system. Coal dust mixed with air is often claimed to be of an explosive nature and many accidents are reported to have resulted from this cause. Many engineers, however, refute this on the basis of experiments which show that explosion can only occur at temperatures high enough to drive off the volatile gases.*

37. Rate of Combustion with Powdered Fuel. — In forcing large quantities of dust into the furnace the velocity imparted to the particles may be so great as to carry them beyond the zone of combustion before oxidation is complete, with the result that the flues, and the back of the furnace, become covered with unconsumed carbon. So much depends upon the depth of the furnace and the arrangement of the regenerative surface that no specific figures can be given as to the maximum rate of combustion that can be efficiently effected. At ordinary rates of combustion the small particles of fuel are completely oxidized while in the combustion chamber and there is total absence of smoke. The use of anthracite coal is practically excluded from this type of stoker unless mixed with coal high in volatile matter. This is due to the fact that fixed carbon burns more slowly than the hydrocarbon gases and the temperature of ignition is higher, hence the most gentle draft will carry away the particles before they are completely consumed. With fuels high in volatile matter the hydrocarbons are distilled at a comparatively low temperature, forming an inflammable gas which burns rapidly with the fixed carbon. A mixture of 30 per cent bituminous and 70 per cent anthracite has been successfully burned in the powdered form.

38. Cost of Pulverizing Coal. — In stokers of the "Aero Pulverizer" type in which the grinding and feeding are carried on simultaneously in a self-contained apparatus, the power consumed varies from 2 per cent to 10 per cent of the total power developed, depending upon the nature of the fuel, the load factor, the efficiency of the driving mechanism, and the degree of fineness of the powdered fuel; 5 per cent is a fair average. The best results are obtained when 95 per cent of the dust will pass a 100 mesh and 75 per cent a 200 mesh, though satisfactory results have been obtained with as low as 40 mesh. Powdered

* See *Fuel*, Jan. 12, 1909, p. 294.

coal in the open market ranges from 25 cents to 50 cents a ton above the price of the same coal in the form of screenings.

39. Efficiency of Powdered-Coal Furnaces. — Table 7 gives the results of a comparative test of a 140-horse-power Babcock & Wilcox boiler, hand fired, *vs.* coal-dust feeder. The test was conducted by the engineering staff of the McCormick Harvester Company at Chicago, Illinois, and the results were obtained with boilers working under normal conditions. The dust apparatus was a modified " Ruhl " feeder, and was installed by the C. O. Bartlett & Snow Company of Cleveland, Ohio. In this particular test the efficiency of the boiler is very low for both hand fired and dust-feed, but the dust-feed test shows an efficiency of 10 per cent greater than that of the hand fired.

TABLE 7.

COMPARATIVE TEST OF 140-HORSE-POWER BABCOCK & WILCOX BOILER.

Hand Fired *vs.* Pulverized Fuel.

Boilers fired by	machine	hand
Date	2-24-04	2-3-04
Duration of test	8 hours	8 hours
Total water evap. into dry steam from and at 212 degrees	70,070	45,673
Average gauge pressure	79.8 lb.	79.4 lb.
Average feed-water temperature, Fahr.	169.7	172.1
Average stack-gas temperature, Fahr.	506	458
Kind of coal used	Westville, Indiana, screenings pulverized to 40 mesh	Westville screenings.
Cost of coal delivered in boiler room ready to fire	$2.10	$1.72
Total weight of dry coal consumed	9,373	8,413
Per cent of ash in coal determined by laboratory analysis	17.5	19.54
Per cent of ash as removed from ash pit and furnace	none	20.57
Heating value of coal	12,555	11,300
Water evap. per pound of fuel, actual conditions	6.822 lb.	4.595 lb.
Equivalent water evap. from and at 212 degrees per pound of dry fuel	7.476 lb.	5.429 lb.
Equivalent water evap. per pound combustible	9.132 lb.	6.941 lb.
Horse power developed	254	165.5
Dry fuel per hour per square foot grate surface	19.27	18.65
Equivalent water evap. per hour per square foot heating surface	3.128	2.039
Cost per 1,000 pounds water evaporated (for fuel ready to fire only)	$0.1455	$0.177
Efficiency of boiler and furnace based on coal	55.5 per cent	41.5 per cent

A comparison of a number of tests of hand-fired and powdered-coal furnaces with different types of feeders shows a decided gain in efficiency of the powdered coal over the hand-fired where the fuel is of a low grade. The gain becomes less marked with fuel of fair quality and disappears entirely with good fuel and properly manipulated automatic stokers. A test made by G. H. Barrus on a 250-horse-power B. &. W. boiler at the General Electric Works in connection with a coal-dust feeder manufactured by the Phœnix Investment Company of New York gave a boiler and furnace efficiency of 75.3 per cent. Subtracting from this the power consumption of 5 per cent for operating the crusher and feeder, the net efficiency was 70.1 per cent. A test of a 135-horse-power return tubular boiler with this same stoker gave a combined efficiency of boiler and furnace of 80 per cent. These figures, however, have been equaled and even exceeded in special hand-fired automatic stoker tests, and only a comparative test of the two systems under similar conditions will show their respective efficiencies.

Tests of Pulverized Fuel: Engr. U.S., April 1, 1904; Engr. Lond., Jan. 31, May, 1904; Power, May, 1904. *Comparative Boiler Tests with Ordinary and Pulverized Coal Firing:* Eng. Rec., March 12, 1904.

40. Furnaces for Burning Powdered Coal. — In burning ordinary bulk coal the mass of incandescent fuel stores up a sufficient quantity of heat to effect the distillation and ignition of the volatile matter in the green fuel. With pulverized coal a refractory lining is necessary to bring about the same result. In arranging a furnace for burning powdered coal in connection with a burner of the forced draft type, the grate bars are removed, ash and fire doors bricked up, and the nozzle bricked in tightly. The lower surfaces of the tubes are covered and the whole forms a reverberatory furnace. With the natural draft system of burner, a suitable opening is left in the brick lining of the ash door to allow the necessary amount of air for combustion to enter. Considerable difficulty is found with delivery nozzles in the formation of slag in the outlet and in their rapid destruction on account of the intense heat. A water-jacketed cast-iron nozzle is said to satisfactorily overcome these objections.

41. Draft for Powdered Fuel. — A study of a number of tests of boilers burning powdered coal shows that the necessary draft is very low and ranges from 0.05 to 0.2 of an inch of water and averages not far from 0.1 inch.

42. Types of Powdered-Coal Burners. — Powdered-coal burners may be grouped into two general classes:

1. The dust-feed burner, in which the coal is supplied in the powdered form, and

2. The self-contained burner, in which the coal is crushed, pulverized, and fed to the furnace simultaneously.

The dust may be fed into the furnace by

1. Natural draft.
2. Mechanical means, or by
3. Forced draft.

The following outline gives a classification of a few of the best known coal-dust burners:

Natural Draft	Natural Draft Feed	Pinther, Wegener	Dust Feed
	Brush Feed	Schwartzkopff	Dust Feed
Forced Draft	Blower Feed	Cyclone, Triumph	Dust Feed
	Compressed Air	Eng. and Powdered Fuel Company	Self-contained
	Paddle Wheel	Ideal, Aero-Pulverizer	Self-contained

43. Pinther Apparatus. — Fig. 8 shows a section through a Pinther coal-dust feeder, illustrating the principles of the "natural draft feed" type. The powdered coal is placed into hopper *B*, from which it is fed by rollers *a*, *a* into the chamber leading to the furnace *C*. The dust falls in a thin stream and is caught up by the current of air and drawn into the furnace as indicated. The furnace is lined with refractory material heated to a sufficiently high temperature to ignite the fuel and burn it in suspension. The chief drawback to a burner of this type is its limited capacity. Any attempt to feed large quantities of fuel into the furnace necessitates such a strong current of air as to carry the particles of dust beyond the zone of combustion before they are completely consumed. Within the limits of its capacity it is an efficient and simple apparatus, but is open to the same objection as all

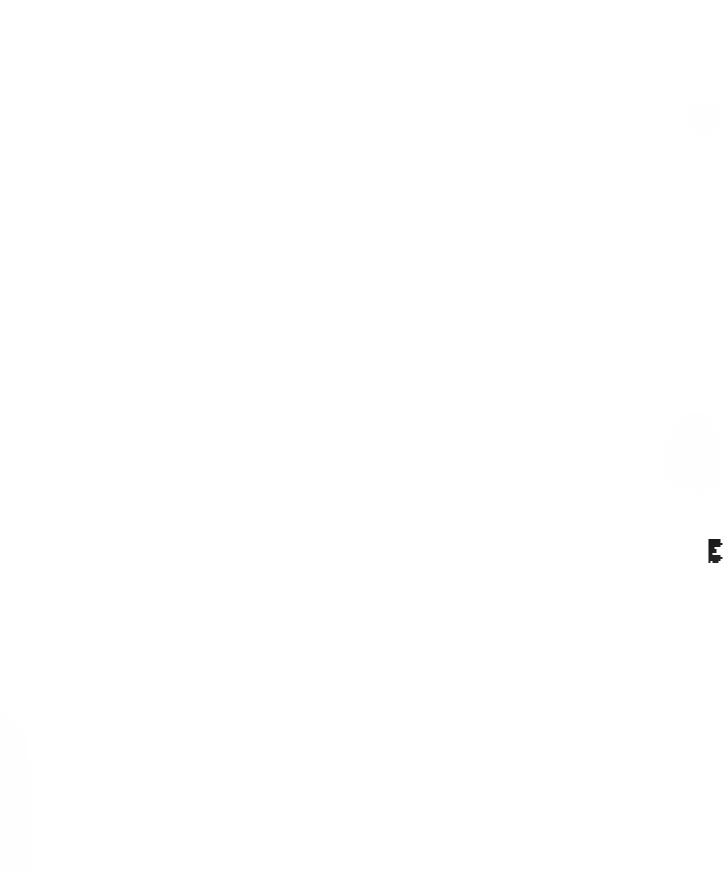


FIG. 8. Pinther Coal-dust Feeder.

burners of this type in that it necessitates the storage of powdered coal. This apparatus is not much in evidence in boiler plants.

44. Schwartzkopff Apparatus. — Fig. 9 shows a section through a Schwartzkopff feeder, illustrating the principles of the brush-feed, natural-draft system. It is a very simple and practical dust feeder, though open to the objection of all systems which require the coal to be ground and pulverized in separate machines. The fuel is placed in a hopper and its supply to the brush is regulated by the hand screw *A* and the spring plate bottom of the hopper. The brush, consisting of a number of flat steel leaves $\frac{1}{32}$ inch by $\frac{1}{8}$ inch wide, revolves at a high speed, 1000 to 1200 r.p.m. and forces the dust into the furnace. The air for combustion is admitted either through the grates in the ordinary way or through the lower chamber of the burner. To prevent the dust from bridging in the hopper, a small hammer *C* is fitted to the brush so that it will strike the plate *D* and agitate the dust. This apparatus is meeting with much success in connection with annealing furnaces, but is still in the experimental state as far as boiler firing is concerned.

FIG. 9. Schwartzkopff Coal-dust Feeder.

45. Aero-Pulverizer Apparatus. — Fig. 10 gives a general view of the Aero-Pulverizer Company's apparatus, and is a typical example of a self-contained system. It is very compact, occupying a floor space of only 30 by 77 inches, and is capable of burning 300 to 1500 pounds of coal per hour. It consists essentially of four interior communicating chambers of successively increased diameter in which paddles revolve on arms with corresponding increased radii. The largest chamber contains a fan, the function of which is to draw the pulverized material successively from one chamber to another and to finally deliver it through the exit in the fan chamber under the impetus of a forced draft. There are two adjustable inlets for air at the feed of the machine through which is introduced the amount of air required for pulverizing purposes. The apparatus may be belt driven or direct connected and runs at about

1600 r.p.m., requiring from 6 to 15 horse power for its operation. It is a complete dust fuel feeding system on one bed plate comprising a pulverizer, fan, coal feeder, hopper, and air dampers. The operation is as follows: Coal previously crushed to nut size is fed to the hopper from

Fig. 10. Aero-Pulverizer Coal-dust Feeder.

the bottom of which it is transferred, with the necessary air for combustion, to the pulverizer chamber. The coal, passing into the pulverizer, is thrown out radially by centrifugal force, due to the rapidly revolving arms and bats, and is reduced to a dust by percussion and attrition. The dust is moved over the ends of the bats and into the fan chamber from which it is blown into the furnace. This apparatus will successfully pulverize and feed coal containing as much as 10 per cent moisture.

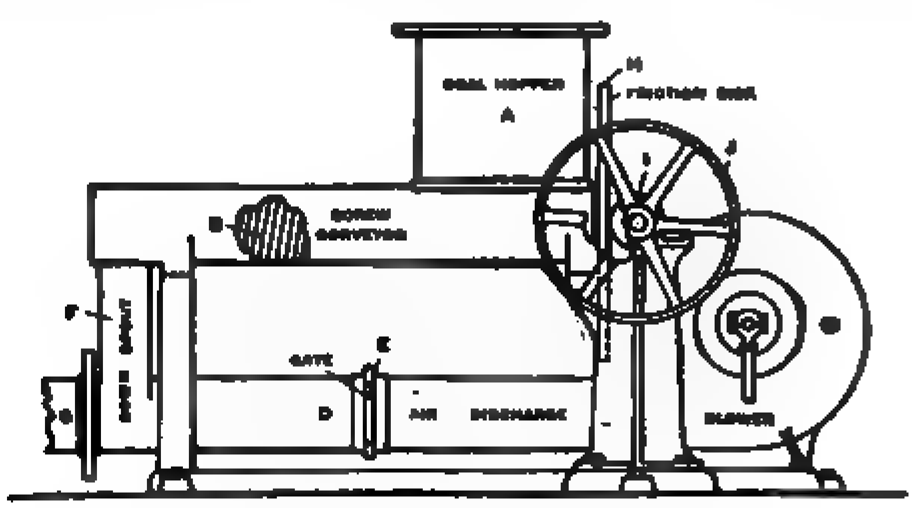


Fig. 11. Triumph Coal-dust Feeder.

46. Triumph Apparatus.—Fig. 11 illustrates the Triumph coal-dust feeder as designed by the C. O. Bartlett & Snow Company, Cleveland, Ohio.

The coal is fed from storage bin to hopper A and feed worm B. The latter forces it down spout F directly to delivery tube D, where it is caught by the air draft and fed into the furnace.

The amount of feed depends upon the speed of the feed worm, which is driven by the friction disk I against the flange plate H. This disk is moved in or out by handle so as to get any speed desired. The air is furnished by fan C, the amount being controlled by valve E.

DESCRIPTION OF COAL-DUST BURNERS.

Aero-Pulverizer System: Eng. News, Nov. 28, 1901, p. 415; Eng. Rec., May 25, 1901, p. 506; Power, March, 1904.

Cyclone System: Engr. U.S., April 1, 1903, p. 272; Eng. News, Nov. 28, 1901, p. 415; Power, March, 1904.

Davis Pulverizer: Jour. Asso. Eng. Soc., July, 1903; Engr. U.S., April 1, 1903.

Ideal: Am. Elecn., April, 1902, p. 196; Power, March, 1904.

Miscellaneous Coal Dust Burners: Am. Elecn., Sept., 1901, p. 434; Engr., Lond., Sept. 11, 1896; Engng., Jan. 15, 1897; Power, Aug., 1903; St. Ry. Rev., Vol. 8–187, 1898; Engr. U.S., April 1, 1904.

Rowe: Engr. U.S., Jan. 1, 1903, p. 93; April 1, 1903, p. 272; Eng. News, Dec. 25, 1902, p. 548; Eng. Rec., Dec. 20, 1902, p. 591.

Schwartzkopff: Am. Elecn., Jan., 1902; Eng. News, Feb. 20, 1902; Power, March, 1904.

Wegener: Cassier's, March, 1896, p. 501; Power, March, 1904; Eng. Mag., March, 1896, p. 1158, Aug., 1896, p. 964, Oct., 1898, p. 125; Eng. News, Sept. 16, 1897, p. 189.

47. Fuel Oil. — The recent development of oil wells in the Western and Gulf States, with the consequent enormous increase in production, has given a marked impulse to the use of crude oil for fuel purposes in steam power plants. Where economic and commercial conditions permit, it is the most desirable substitute for coal. The total absence of smoke and ashes, prompt kindling and extinguishing of fires, extreme rate of combustion, and ease with which it can be handled and controlled are marked advantages in favor of fuel oil. The reduction in volume and weight over an equivalent quantity of coal for equal heating values and the increase in boiler efficiency are factors of no mean importance, particularly in connection with marine or locomotive work. In stationary work the chief objections are the difficulty in securing ample storage capacity and the increased rate of insurance. An objection sometimes raised against oil fuel is the increased depreciation of the setting, but in a well-designed setting this figure is only nominal and of secondary importance. However, in spite of the many advantages presented in the use of fuel oil for power plant purposes, the limited supply and constant fluctuation in price prevent its adoption as a general fuel, and limit its use to the plants most favorably located.

Crude Oil Burning: Power, March, 1907; Engr. U.S., Dec. 15, 1905, March 1, 1906; Am. Elecn., Aug., 1903, p. 396; Engng., March 28, 1902, p. 140; Eng. Mag., May, July, Sept., 1903; Eng. News, June 19, 1902, p. 501; Cassier's, May, 1901, p. 61; Engr., Lond., Dec. 9, 1904; St. Ry. Jour., May 10, 1902, p. 588; Am. Gas Light Jour., May 12, 1902, p. 695.

TABLE 8.

ANALYSES OF TYPICAL AMERICAN FUEL OILS.

Location.	Authority.	Physical Properties.				
		Specific Gravity. 60°–70° F.	Flash Point. Deg. F.	Burning Point. Deg. F.	Specific Viscosity.	
					60° F.	185° F.
California — Crude	Ed. O'Neill	0.9533	......	......	299.6	4.7
Do	....do	0.9572	......	......	373	......
Do	....do	0.7825	62	64.5	1.17	......
Do	....do	0.9670	196	221	......	......
Kansas — Crude	B. F. McFarland.	0.866	52	77	......	......
Louisiana — Crude	C. E. Coates	......	......	......	......	......
Ohio — Distillate	Deville	0.887	......	......	......	......
Do	N. W. Lord	0.838	177	212	......	......
Pennsylvania — Crude	Deville	0.826	......		......	......
Pennsylvania—Distillate	....do	0.886	......		......	......
W. Virginia — Crude	....do	0.841	......		......	......
Wyoming — Crude	Colburn	......	......	......	......	......
Texas — Crude	Denton	0.92	142	181	......	......
Texas — Distillate	U. S. Naval Report	0.926	216	240	......	......

Location.	Authority.	Chemical Properties.				
		C	H	O+N	S	B.T.U. per Lb.
California — Crude	Ed. O'Neill	85.75	11.3	......	0.668	18,797
Do	....do	86.3	10.7	......	0.8	18,646
Do	....do	......	......	......	......	......
Do	....do	......	......	......	......	......
Kansas — Crude	B. F. McFarland.	85.4	13.07	......	......	......
Louisiana — Crude	C. E. Coates	......	......	......	0.34	19,814
Ohio — Distillate	Deville	84.2	13.1	2.7	......	18,718
Do	N. W. Lord	......	......	......	......	19,880
Pennsylvania — Crude	Deville	82	14.8	3.2	......	17,930
Pennsylvania—Distillate	....do	84.9	13.7	1.4	......	19,210
W. Virginia — Crude	....do	84.3	14.1	1.6	......	18,400
Wyoming — Crude	Colburn	......	......	......	......	19,590
Texas — Crude	Denton	84.6	10.9	2.87	1.63	19,060
Texas — Distillate	U. S. Naval Report	83.26	12.41	3.83	0.50	19,481

48. Chemical and Physical Properties of Fuel Oil. — From Table 8 it will be seen that the physical properties of oils from different localities in the United States differ widely, while the chemical constituents vary

but slightly. For example, the oils given in the table differ greatly in volatility, specific gravity, and viscosity, but have nearly a constant ratio of carbon and hydrogen and consequently vary but slightly in heating value.

A good deal of the oil produced is unfit for fuel purposes unless refined. The chief impurities are sulphur, earthy matter, and water. Besides interfering with the free burning of the oil, moisture and sulphur have a deleterious effect upon the boiler and furnaces, and should not be present in large quantities. Where the percentage of sulphur is greater than 4 per cent, the depreciation of the boiler and furnace offsets the gain in using the lower grade fuel. Many successful processes of removing the water and sulphur are on the market, and consequently crude oil high in sulphur should not be used unless the depreciation element has been taken into consideration.

Oil that is to be transported or stored or used for fuel inside of buildings should be of the "reduced" variety, from which the naphtha and higher illuminating products have been distilled. The gravities of such distillates vary from 20 to 25 degrees Baume, or close to 0.9 specific gravity, and their flash points range from 240 degrees F. to 270 degrees F. This variation in volatility has little effect on the heat value of the oil, since the ratio of carbon to hydrogen varies but slightly in the various distillates.

One barrel of oil contains 42 gallons and weighs from 310 to 332 pounds according to the specific gravity. Compared with coal, oil occupies about 50 per cent less space and is 35 per cent less in weight for equal heat values. The comparative heat values of coal and oil are approximately as follows:

B.T.U. per Pound of Coal.	Pounds of Coal Equal to 1 Barrel of Oil.	Barrels of Oil Equal to 1 Short Ton of Coal.
10,000	620	3.23
11,000	564	3.55
12,000	517	3.87
13,000	477	4.19
14,000	443	4.52
15,000	413	4.84

Technical Aspects of Oil as Fuel: Junge, Power, Oct., 1907, p. 665.

Petroleum Oil Fields: Jour. Soc. Chem. Ind., Oct. 15, 1902, p. 1228. *Investigation on American Petroleum:* Am. Chem. Jour., March, 1896, p. 215. *The Constituents of Pennsylvania, Ohio, and Canada Petroleums:* Amer. Chem. Jour., Vol. 19–419. *Composition of California Petroleum:* Amer. Chem. Jour., Vol. 19–796, Vol. 25–253. *Composition of Petroleum:* Amer. Chem. Jour., Vol. 28–165, 33–251. *Composition*

of Texas Petroleum: Jour. Amer. Chem. Soc., Feb. 9, 1901, p. 264; Soc. Chem. Ind., 19–121, 20–237, 690. *Flashing Points of Petroleum:* Jour. Soc. Chem. Ind., 15–341. *Origin of Petroleum:* Jour. Soc. Chem. Ind., 16–727, 1898. *Influence of Water on Flash Test and Combustion Point of Petroleum:* Chem. News, 85–267. *The Relation between Some Physical Properties of Bitumens and Oils:* Eng. Rec., Aug. 18, 1906.

49. Efficiency of Boilers with Fuel Oil. — From Table 14, it will be seen that 70 per cent is a high figure for boiler efficiency in regular service when good coal is burned, and 65 per cent a fair average. With liquid fuel an average efficiency of 4 to 6 per cent above this is readily attained. (See Table 9.) This increase in efficiency is partly due to the fact that the oil is readily broken up and brought into intimate contact with the necessary air for combustion and the loss due to excess of air is correspondingly reduced. The results of tests made by the Liquid Fuel Board of the U.S. Navy show that oil has an efficiency of 5 per cent greater than coal for the same rate of evaporation, and that the boiler capacity may be increased 50 per cent above that possible when burning coal and still maintain the same efficiency. The maximum efficiency with oil was attained at a higher rate of evaporation than was the maximum efficiency when coal was burned.

TABLE 9.

BOILER EFFICIENCIES, OIL FUEL.

Index.	Authority.	Reference.	Quality of Oil.	Evaporation from and at 212° F. Pounds.	Efficiency of Boiler and Furnace. Per Cent.
1	Pacific Light and Power Co., Los Angeles, Cal.	Eng. Record, Aug. 6, 1904.	California Crude, 18,607 B.T.U. per pound.	16.02	83.06
2			California Crude, 18,760 B.T.U.	15.66	80.6
3	U.S. Naval Board	1902 Report of U.S. Naval Liquid Fuel Board.	Reduced Beaumont, 19,480 B.T.U.	14.43	71.5
4			20,000 B.T.U.....	16.9	77.8
5	Prof. Williston...	Engineering Magazine, July, 1903.	West Va. Crude, 20,960 B.T.U.	16.5	76
6			Texas Crude, 18,850 B.T.U.	15.9	76.8
7	Prof. Denton....	Power, Feb., 1902.	Beaumont Texas, 19,060 B.T.U.	15.5	78.5
8	Wallsend	Engineering, Nov. 6, 1902.	Not stated.......	14.45	....

50. Comparative Evaporative Economy of Oil and Coal. — In determining the comparative economy of coal and oil, the fixed and operating charges must be considered in addition to the cost and efficiency of the fuel. From the market quotation on oil and coal and the comparative heating values of each the actual cost per B.T.U. is readily obtained, and by combining this with the relative efficiencies from the furnace standpoint the net cost of the fuel is obtained. The fixed charges vary with the location and size of the plant and are approximately the same per boiler horse power for a given location in both cases. The insurance rates may be greater with the oil fuel and the depreciation of the boiler setting may be somewhat larger, but in a well-constructed furnace the latter item should be the same in both instances for average rates of combustion. The operating charges are decidedly in favor of the oil fuel, since no ash handling is necessary. Oil fuel is readily fed to the furnace, and the cost of attendance may be materially less than with coal firing, and one man may safely control from eight to ten boilers. Table 106, Chapter XVII, gives data relative to the cost of producing electrical power in connection with oil-fired steam plants.

Tests of Crude Oil as a Fuel: Cassier's, May, 1901, p. 61; Power, Feb., 1902, p. 8; Eng. U.S., Jan. 16, 1905, p. 90, Feb. 1, 1905; Eng. Rec., Dec. 20, 1902; Eng. Mag., July, 1902, p. 615; Eng. News, July 11, 1901, p. 23; Eng. Rec., Aug. 6, 1904, p. 175; Oct. 29, 1904, p. 502.

51. Oil Burners. — The function of the burner is to atomize the oil to as nearly a gaseous state as possible.

Classification of a few well-known burners:

Mechanical Spray:

- Kōrting.

Vapor or Carburettor:

- Durr.
- Harvey.

Spray Burners:

- Outside Mixers.
  - *a.* Booth.
  - *b.* Warren.
- Inside Mixers.
  - *a.* Hammel.
  - *b.* Kirkwood.
  - *c.* Branch.
  - *d.* Williams.

Oil burners for burning liquid fuel may be divided into three general classes:

1. Mechanical spray, in which the oil previously heated to a temperature of about 150 degrees F. is forced under pressure through nozzles so designed as to break it up into a fine spray. The Körting Liquid Fuel Burner, Fig. 12, is an example of this type. In this design a central spindle, spirally grooved, imparts a rotary motion to the oil and causes it to fly into a spray by centrifugal force on issuing from the nozzle. The particles of oil are burned in the furnace when they come in contact with the necessary air to effect combustion. This type of burner is little used in this country in connection with power plant work, but is meeting with much success on the continent.

Fig. 12. Körting Fuel Oil Burner.

2. Vapor burners, or carburettors, in which the oil is volatilized in a heater or chamber and then admitted to the furnace, are seldom used except in connection with refined oils, as the residuals from crude oil are vaporized only at a high temperature. The Durr and Harvey gasifiers are the best known of this type.

3. Spray burners are by far the most common in use. In this type the oil is held in suspension and forced into the furnace by means of a jet of steam or compressed air. Spray burners are designed either as *outside mixers,* in which the oil and atomizing medium meet outside the apparatus, or *inside mixers,* in which the oil and atomizing medium mingle inside the apparatus.

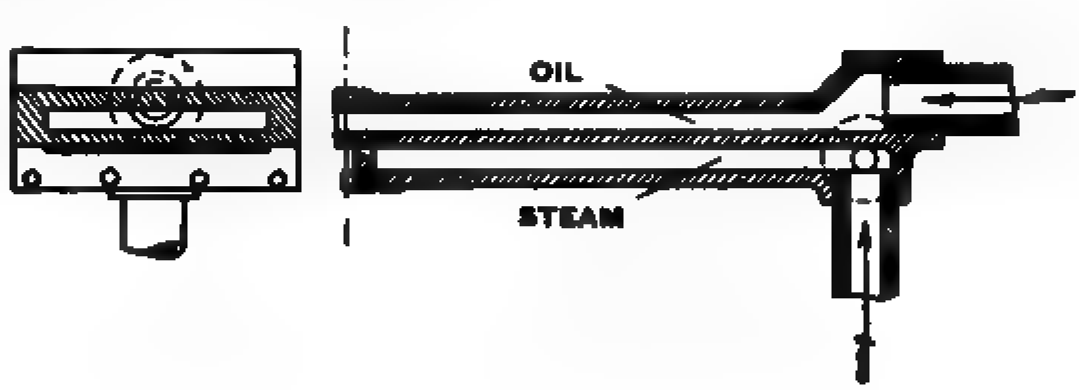


Fig. 13. Booth Fuel Oil Burner.

The *Booth burner*, Fig. 13, illustrates the principles of the "outside mixer" type of apparatus and is in use on the Santa Fe Railroad. In this type the oil flows through a thin slit and falls upon a jet of steam

which atomizes it and forces it into the furnace. A feature of this apparatus is its simplicity of construction and freedom from clogging.

Fig. 14 illustrates the Hammel burner as used at the power house of the Pacific Light and Power Company, Los Angeles, Cal. Oil enters the burner under pressure and flows through opening D to the mouth of the burner where it is atomized by the steam jets issuing from slots G, H, and I. The oil is preheated to facilitate its flow through the supply system. Plates K-K are removable and are easily replaced when worn out or burned. The Hammel burner belongs to the " inside mixers."

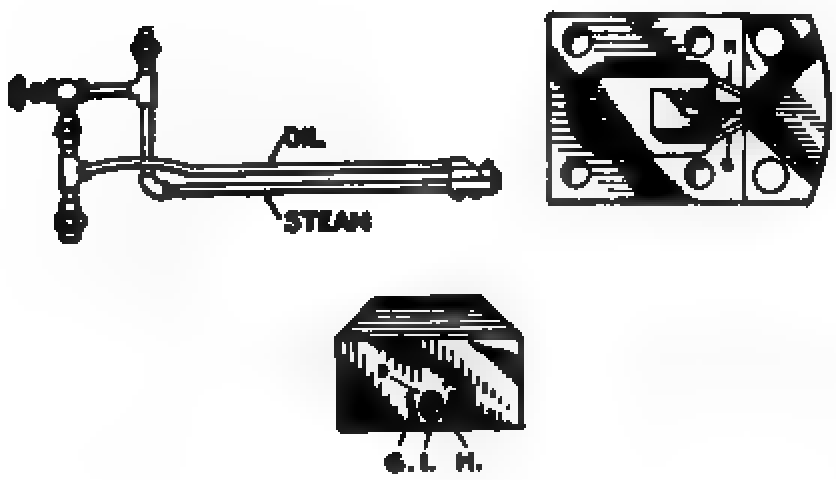


FIG. 14. Hammel Fuel Oil Burner.

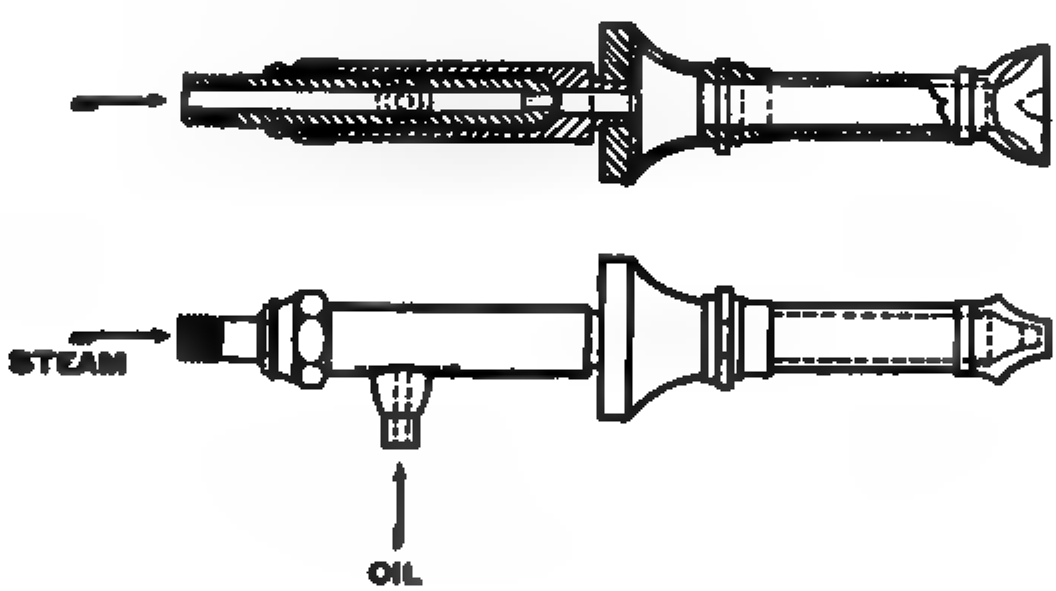


FIG. 15. Branch Fuel Oil Burner.

A few well-known types of "inside mixers" are illustrated in Figs. 14 to 16. The operation is practically the same in all of them and they differ only in mechanical details.

FIG. 16. Kirkwood Fuel Oil Burner.

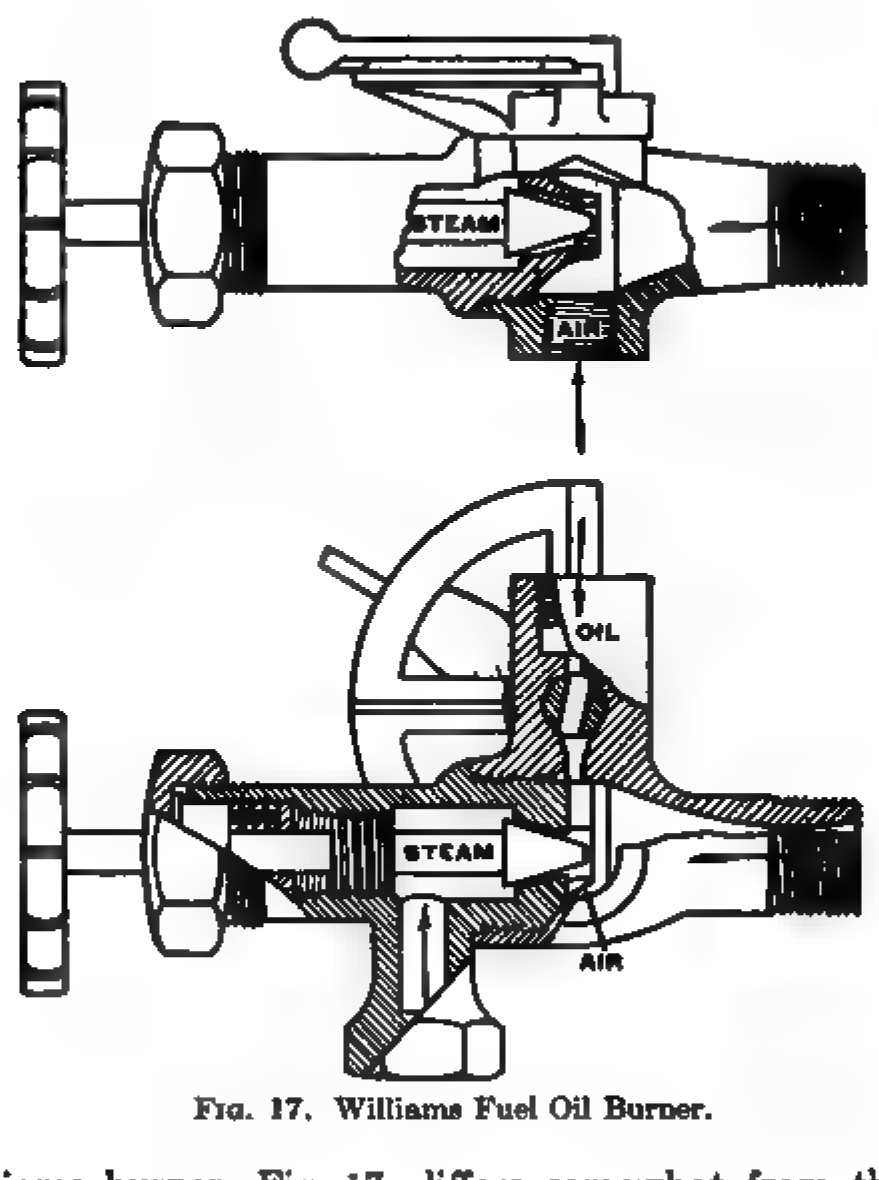


FIG. 17. Williams Fuel Oil Burner.

The Williams burner, Fig. 17, differs somewhat from the others in that the air supply passes through the burner and mingles with the oil and steam before entering the furnace.

The simplest and most reliable burners are of the Hammel type and are much in evidence in the Pacific States.

Notes on Oil Burners using Compressed Air: Power, Nov., 1904. *Report of U.S. Naval Liquid Fuel Board:* Engr. U.S., Dec. 1, 1904. *Oil Burners:* Engng., April 15, 1904; Am. Engr. and R.R. Jour., Sept., 1901.

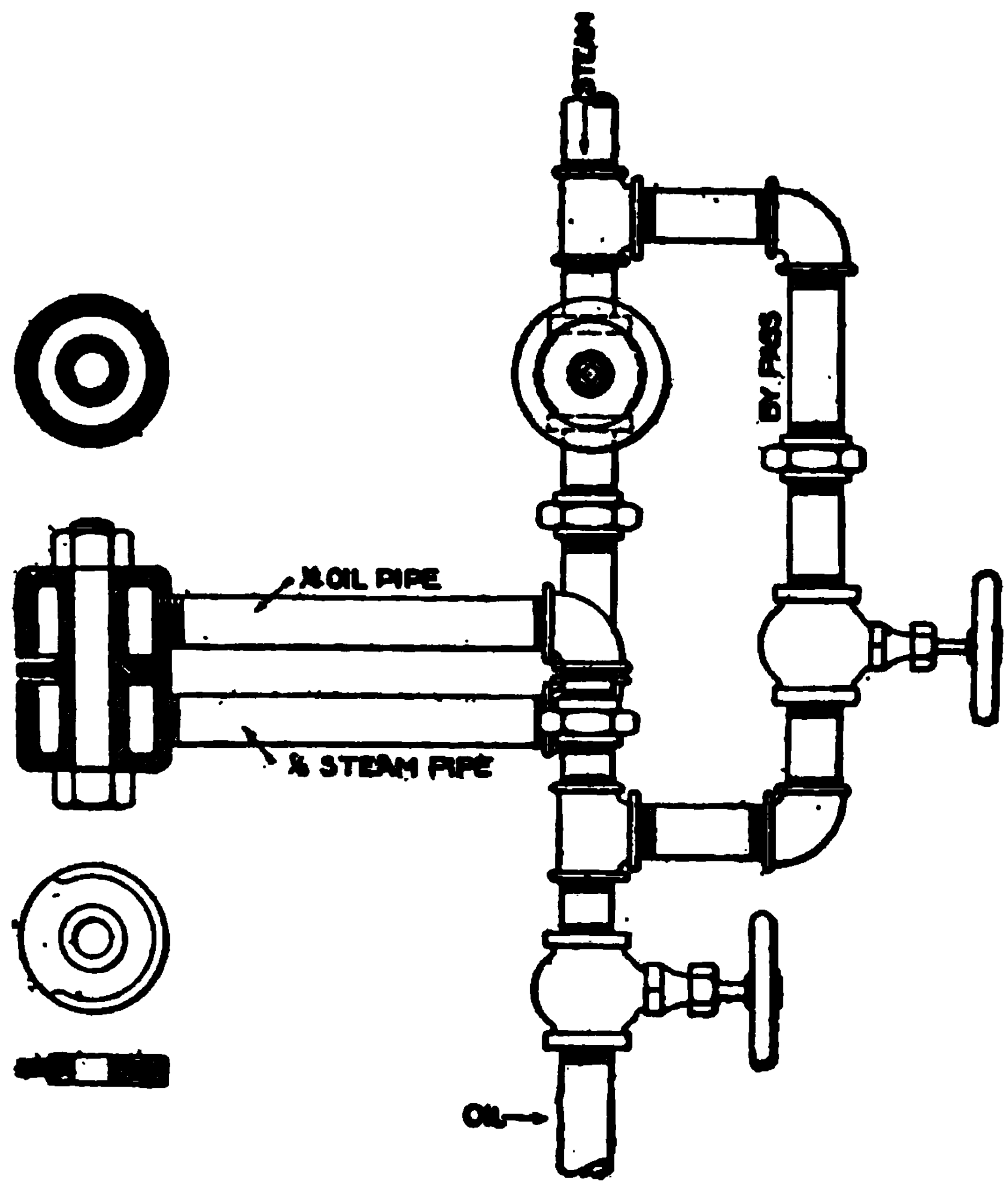


Fig. 18. Warren Fuel Oil Burner.

52. Furnaces for Burning Oil Fuel. — Fig. 19 shows the construction of a typical oil-burning furnace as applied to a 250-horse-power B. & W. water tube boiler in the power plant of the Union Loop Elevated Station, Chicago. For the successful burning of oil the furnace should be so constructed that oxidation of the fuel is complete before it reaches the tubes. This is effected by arranging the refractory lining to form a sort of reverberatory furnace in which the atomized oil is vaporized and mixed with the necessary air for combustion. The air is preheated in its passage beneath the lower lining of the furnace and the supply is regulated by a suitable damper. The regulation of air, steam, and oil for the burner is a very delicate operation and requires considerable skill for efficient results.* In the particular furnace illustrated in Fig. 19 the flame impinges against a cellular wall of fire brick before it reaches

* For a modern and highly efficient system of oil fuel feeding and regulation, see *Power*, Dec. 29, 1908, p. 1108.

the bridge wall. The bricks are loosely stacked and are readily removed when burned out. They tend to save the lining of the bridge wall arch and insure a more intimate mixture of air and oil in the combustion chamber. The supply of crude oil is limited compared with that of coal and the price is subject to sharp fluctuations, and consequently the cost may prove prohibitive for fuel purposes. To be prepared for such an emergency, many engineers design the furnaces for coal burning and arrange them with loose brick-work for the temporary use of oil.

FIG. 19. Furnace for burning Fuel Oil, Front Feed.

Fig. 20 illustrates the application of a Hammel burner at the rear end of a furnace.

Oil-Fired Furnaces: Engr. U.S., July 1, 1902, p. 491, Nov. 15, 1905; Eng. Mag., May, 1903; Engng., April 15, 1904, p. 523, April 29, 1904, p. 594. *Liquid Fuel Combustion:* Trans. A.S.M.E., May, 1902. *Equipment for Oil Fuel:* Eng. and Min. Jour., Oct. 7, 1905.

53. Air vs. Steam as an Atomizing Medium. — Table 10 gives the results of a series of tests made by the U.S. Naval Liquid Fuel Board in 1902 on different types of burners using air, steam, or both for atomizing the fuel. The first eight tests were made with compressed air as the spraying medium and under pressures varying from 0.78 pounds to 4.68 pounds per square inch. The most economical results were

obtained with the lower pressures, and the total steam used to compress the air varied from 1.06 per cent to 7.45 per cent of the total steam generated, but not all burners would work with air at this low pressure. With steam as the spraying medium the steam required to

Fig. 20. Furnace for burning Fuel Oil, Rear Feed.

operate the burner varied from 3.98 per cent to 5.77 per cent of the total generated, while the burner using both steam and air required 6.09 per cent to 8.54 per cent of the total. The results of recent tests with the latest types of burners give somewhat lower steam consumption than the tests conducted by the Naval Board and a good average is not far from 3 per cent.* Table 10 contains also the results of a few scattering tests made with different types of burners.

In general it may be said that where a supply of low-pressure air is available, air is unquestionably more economical than steam as an atomizing medium, but in the average boiler plant the use of steam obviates complication and risk of interrupted service. Where it is necessary to use high-pressure air the economy decreases with the increase in pressure, since the cost of each cubic foot of compressed air increases rapidly with the pressure, but its ability to atomize the oil does not increase proportionately.

Steam vs. Air for Liquid Fuel and Oil Burners: Am. Mach., Vol. 27, No. 51.

* See *Proceedings*, A. S. M. E., Dec., 1906, p. 1698.

TABLE 10.

TESTS OF FUEL-OIL BURNERS.

Number of Test.	Authority of Test.	Reference.	Type of Burner.	Steam Pressure (Pounds per Square Inch Gauge).	Pressure of Medium used for Spraying Oil.	Temperature of Spraying Medium (Degrees F.).	Equivalent Evaporation from and at 212° F. (Pounds).	Steam used in Spraying Oil; per cent of Total Steam generated.	Efficiency of Boiler (per cent).
1			O. C. B. W. (Air)	273.5	3.2	......	12.7	2.38	62.8
2		Report of U.S. Naval Liquid Fuel Board. 1902.	......do	273.5	4.62	......	12.18	3.89	60.3
3			......do	273.5	0.78	102.5	14.43	1.06	71.5
4			......do	273.5	3.37	122	11.73	2.88	58.1
5		Engineer U.S., June 1, 1903.	......do	273.5	1.41	120	14.22	1.97	70.4
6			......do	271.5	1.31	113.5	14.12	1.53	69.9
7	United States Naval Board.	Engineering Record, December 20, 1902.	......do	272.5	4.66	161	13.29	7.45	65.8
8			......do	276	4.68	136	10.77	4.25	53.4
9		Western Electrician, December 13, 1902.	Hayes (Steam)	273.5	32	......	13.89	5.77	68.9
10			O. C. B. W. (Steam)	273.1	29.9	444	13.47	3.98	66.7
11		Engineer, London, Nov. 14, 1902.	......do	273.7	61.4	408	13.45	4.41	66.7
12			......do	274.2	81	401	13.58	5.03	67.3
13			Reed (Air and Steam)	276.7	92	375	14.35	8.54	71.1
14			......do	277.4	89	416	14.06	6.09	69.7
15	Wallsend	Engng., Nov. 6, 1903	Körting *	113	75	240‡	14.45	......	......
16	Denton	Power, February, 1902	Williams (Steam)	86.5	......	......	15.49	3.5	78.5
17	Pacific Light and Power Co. Los Angeles, Cal.	Tests by B. & W. and Stirling Companys' Committee	H. & M. (Steam)	156	36 †	109‡	15.66	2.33	80.6
18			Hammel (Steam)	156	35 †	88‡	14.87	2.72	70.7

* Uses no steam or air, Oil under pressure. † Oil pressure. ‡ Temperature of oil.

54. Oil Pressure. — This varies with the different types of burners and ranges from a few pounds to 60 pounds or more per square inch. The low-pressure systems are ordinarily operated under standpipe pressures as in Fig. 21, which illustrates the arrangement of apparatus as advocated by the International Gas and Fuel Company. A steam pump *B* draws the oil from the buried tank through pipe *Z* and delivers it to the standpipe *E*. Thence it flows through pipe *I* to the burners under a

Fig. 21. International Gas and Fuel Company's Fuel Oil System.

head of about 10 feet. The pump runs constantly, the surplus oil flowing back to the tank through the pipe *T*. The oil is heated by the exhaust pipe *Z′*. The oil pump is provided with a device *D* having a piston connected by a chain with a cock *S*, which automatically opens when the boiler is not under steam pressure, so that the standpipe will be emptied, the oil flowing to the storage tank.

The high-pressure systems are invariably operated by steam pumps, usually in duplicate, and are so arranged that the oil pressure will be kept practically constant irrespective of the steam pressure. The adjustment of steam and oil is a very delicate operation, and fluctuation in the steam pressure disturbs the proportion of oil and steam; to prevent this

the steam pressure at the burner is reduced several pounds below that of the boiler by suitable reducing valves and is thereby kept at a nearly constant value.

55. Oil Storage and Transportation. — Distillates or reduced oils are readily stored and transported, but the crude oils, on account of the inflammability of the highly volatile elements, offer a different problem. In most cities distillates may be stored in large quantities but only in tanks sunk below the lowest level of the surrounding territory. This is a protection against flooding the district with burning oil in case of a fire. In the country the oil is ordinarily stored in tanks above the ground level and at some distance from the plant.

Fig. 22 illustrates the Hydraulic Oil Storage Company's system of storing oil and delivering it to the burners. The oil reservoirs are placed

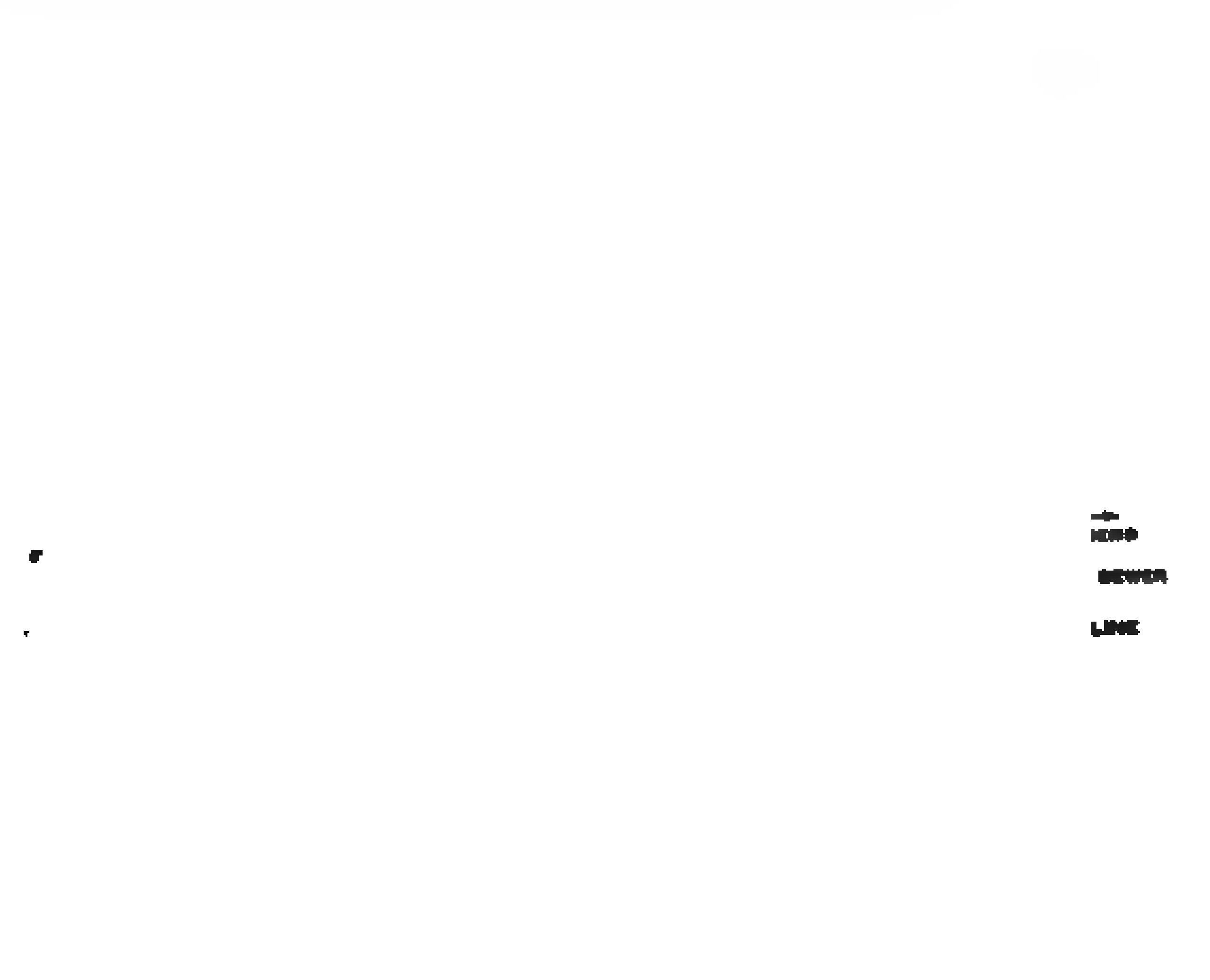


FIG. 22. Hydraulic Oil Storage Company's Fuel Oil System.

below grade as indicated to minimize fire risk. The operation is as follows: Water enters the "float box" and flows through a "three-way cock" to the bottom of the reservoir until all of the oil and water pipes are filled up to the level of the float box, when the float automatically cuts off the supply. This flooding of the entire system

drives out all of the air. The three-way cock is then turned to "discharge" and part of the water flows to the sewer. The tank-car or wagon is next attached to the "oil inlet" and the oil flows into the tank and displaces the water until the level of the "filler float" is reached, when the supply is automatically cut off. The inlet is so placed that the head of oil in the tank-car is sufficiently great to overcome the opposing head of water. The three-way valve is next turned to the first position and the head of water forces the oil to the burners. After the oil has been withdrawn from the storage tank the water can only rise to the level of the water in the float box and therefore cannot be fed to the furnace. The small steam pipe admits steam into the tank and heats the oil, thereby making it flow more freely.

Storing Oil Fuels: Eng. News, Oct. 29, 1903, p. 396.
Petroleum Reservoirs: Jour. Soc. Chem. Ind., Jan. 31, 1899.
Handling Fuel at Railway Terminals: Eng. News, Sept. 25, 1902, p. 232.

56. Conclusions of U. S. Naval Liquid Fuel Board. — After a series of elaborate tests it was concluded

a. That oil can be burned in a very uniform manner.

b. That the evaporative efficiency of nearly every kind of oil per pound of combustible is probably the same. While the crude oil may be rich in hydrocarbons, it also contains sulphur, so that, after refining, the distilled oil has probably the same calorific value as the crude product.

c. That a marine steam generator can be forced to even as high a degree with oil as with coal.

d. That up to the present time no ill effects have been shown upon the boiler.

e. That the firemen are disposed to favor oil, and therefore no impediment will be met in this respect.

f. That the air requisite for combustion should be heated if possible before entering the furnace. Such action undoubtedly assists the gasification of the oil product.

g. That the oil should be heated, so that it could be atomized more readily.

h. That when using steam higher pressures are undoubtedly more advantageous than lower pressures for atomizing the oil.

i. That under heavy forced draft conditions, and particularly when steam is used, the Board has not yet found it possible to prevent smoke from issuing from the stack, although all connected with the tests

made special efforts to secure complete combustion. Particularly for naval purposes, it is desirable that the smoke nuisance be eradicated in order that the presence of a war ship might not be detected from this cause. As there has been a tendency of late to force the boilers of industrial plants, the inability to prevent the smoke nuisance under forced-draft conditions may have an important influence upon the increased use of liquid fuel.

j. That the consumption of liquid fuel cannot probably be forced to as great an extent with steam as the atomizing agent as when compressed air is used for this purpose. This is probably due to the fact that the air used for atomizing purposes, after entering the furnace, supplies oxygen for the combustible, while in the case of steam the rarefied vapor simply displaces air that is needed to complete combustion.

k. That the efficiency of oil-fuel plants will be greatly dependent upon the general character of the installation of auxiliaries and fittings, and therefore the work should be intrusted only to those who have given careful study to the matter and who have had extended experience in burning the crude product. The form of the furnace will play a very small part in increasing the use of crude petroleum. The method and character of the installation will count for much, but where burners are simple in design and are constructed in accordance with scientific principles there will be very little difference in their efficiency. Consumers should principally see that they do not purchase appliances that have been untried and have been designed by persons who have had but limited experience in operating oil devices.

57. Gaseous Fuels. — These fuels offer all of the advantages of liquid fuels and but few of the disadvantages. The gases most commonly met with in connection with steam power plants are outlined in Table 11. The artificial gases for steam purposes are prohibitive in cost in most cases, and even in blast-furnace installations, where the gases are waste products, the gas engine has virtually supplanted the steam engine for power purposes. In the immediate locality of natural gas wells gas-fired furnaces may prove to be more economical than coal furnaces, but the limited supply and constant fluctuation in price limit its use as a general fuel. From the market quotations on coal and gas and the comparative heating value of each the actual cost per B.T.U. is readily obtained, and by combining this with the relative efficiencies from the furnace standpoint the net cost of the fuel is obtained. The following table, based upon the assumption that one cubic foot of natural gas under standard conditions has a heating value of 1,000 B.T.U., will enable an approximate comparison to be made:

B.T.U. per Pound of Coal.	Pounds of Coal Equal to 1,000 Cu. Ft. of Gas.	No. of 1,000 Cu. Ft. of Gas Equal to One Short Ton of Coal.
10,000	100	20
11,000	91	22
12,000	83	24
13,000	77	26
14,000	71	28
15,000	67	30

Fuel Economy: Fuel Economy in Steam Power Plants: Cassier's Mag., May, 1904; Inst. of Elec. Engrs., Jan. 12, 1905; Engr. U.S., April 1, 1905; Engng., Aug. 7, 1903; Eng. Mag., June, 1907. *The Province of the Fuel Expert:* Eng. and Min. Jour., May 25, 1905. *A Gas-Fired Boiler:* Engr. U.S., Feb. 15, 1907, p. 223.

See also A.S.M.E. Code for conducting Boiler Tests — reprinted in Appendix B.

TABLE 11.

CHARACTERISTICS OF GASEOUS FUELS.

(Lucke.)

	H	CO	CH_4	C_2H_4	O
Natural gas	1.7	0.55	94.16	0.30	0.30
Cannel-coal gas	27.7	6.8	50.0	13.0	....
Common-coal gas	39.78	7.04	45.16	6.38	0.06
Carburetted water-gas	21.8	28.1	30.7	12.9	0.5
Uncarburetted water-gas	49.50	35.93	1.05	....	....
Producer-gas, little steam	9.2	25.3	3.1	0.8	....
Loomis Pettibone coal gas	14.0	20.0	2.0	0.20	0.10
Dowson gas, average	18.0	25.0	3.0	....	....
Taylor gas, average	21.0	12.0	2.0	....	3.0
Mond gas	29.0	12.0	2.0	....	....
Coke-oven gas	53.0	6.0	35.0	2.0	....
Blast-furnace gas	3.0	27.5	....	....	....

	CO_2	N	Cubic Feet of Air per Cubic Foot of Gas.	B.T.U. per Cubic Foot of Gas. High.	B.T.U. per Cubic Foot of Gas. Low.
Natural gas	0.29	2.80	9.13	989	888
Cannel-coal gas	0.1	2.4	6.50	843	762
Common-coal gas	1.08	0.50	6.38	727	651
Carburetted water-gas	3.8	2.2	6.00	702	635
Uncarburetted water-gas	4.25	8.75	2.10	295	265
Producer-gas, little steam	3.4	58.2	1.24	160	150
Loomis Pettibone coal gas	8.2	55.5	....	....	....
Dowson gas, average	7.0	47.0	1.32	119	115
Taylor gas, average	6.0	57.0	0.98	130	116
Mond gas	14.5	42.5	1.17	156	139
Coke-oven gas	2.0	2.0	5.06	620	524
Blast-furnace gas	10.0	59.4	.81	100	99

CHAPTER III.

BOILERS.

58. As affecting fuel economy the boiler equipment is by far the most important part of the power plant and involves the largest share of the operating expenses. It matters little how elaborate, modern, or well designed it may be, skill, good judgment, and continued vigilance are required on the part of the operator to secure the best efficiency.

Of the various types and grades of boilers on the market experience shows that most of them are capable of practically the same evaporation per pound of coal, provided they are designed with the same proportions of heating and grate surface and are operated under similar conditions. They differ, however, with respect to space occupied, weight, capacity, first cost, and adaptability to particular conditions of operation and location.

59. Classification. — As to design and construction there is an almost endless variety of boilers and furnaces, classified as *internally* and *externally fired; water tube* and *fire tube; through tube* and *return tubular; horizontal* and *vertical.*

The internally fired type includes the *vertical tubular, locomotive, Scotch-marine,* and practically all *flue boilers.* The externally fired includes the *plain cylinder,* the *through tubular, return tubular,* and nearly all stationary *water-tube* boilers.

60. Vertical Tubular Boilers. — Vertical tubular boilers, Figs. 1 and 23, are commonly used where small power, compactness, low first cost, and sometimes portability are the chief requirements, though they are not necessarily restricted to small sizes. The tubes are sometimes arranged so that the spaces between them radiate from a hand hole on one side so that a scraper may readily be inserted to clean the top of the furnace plate. The hand hole in the water leg permits removal of the scale. It is convenient to place a chain in the bottom of the water leg which can be worked around through the hand hole for the purpose of loosening up the scale deposit. The distance between the furnace crown and top of the grate is never less than 24 inches even in the smallest boiler and should be as great as possible to insure good combustion. Two styles of vertical boilers are in common use, the ordinary vertical

type, Fig. 1, and the submerged type, Fig. 23. In the former the upper tube sheet and part of the tubes are above the water line, and while this feature may tend to superheat the steam to a slight extent, the difficulty

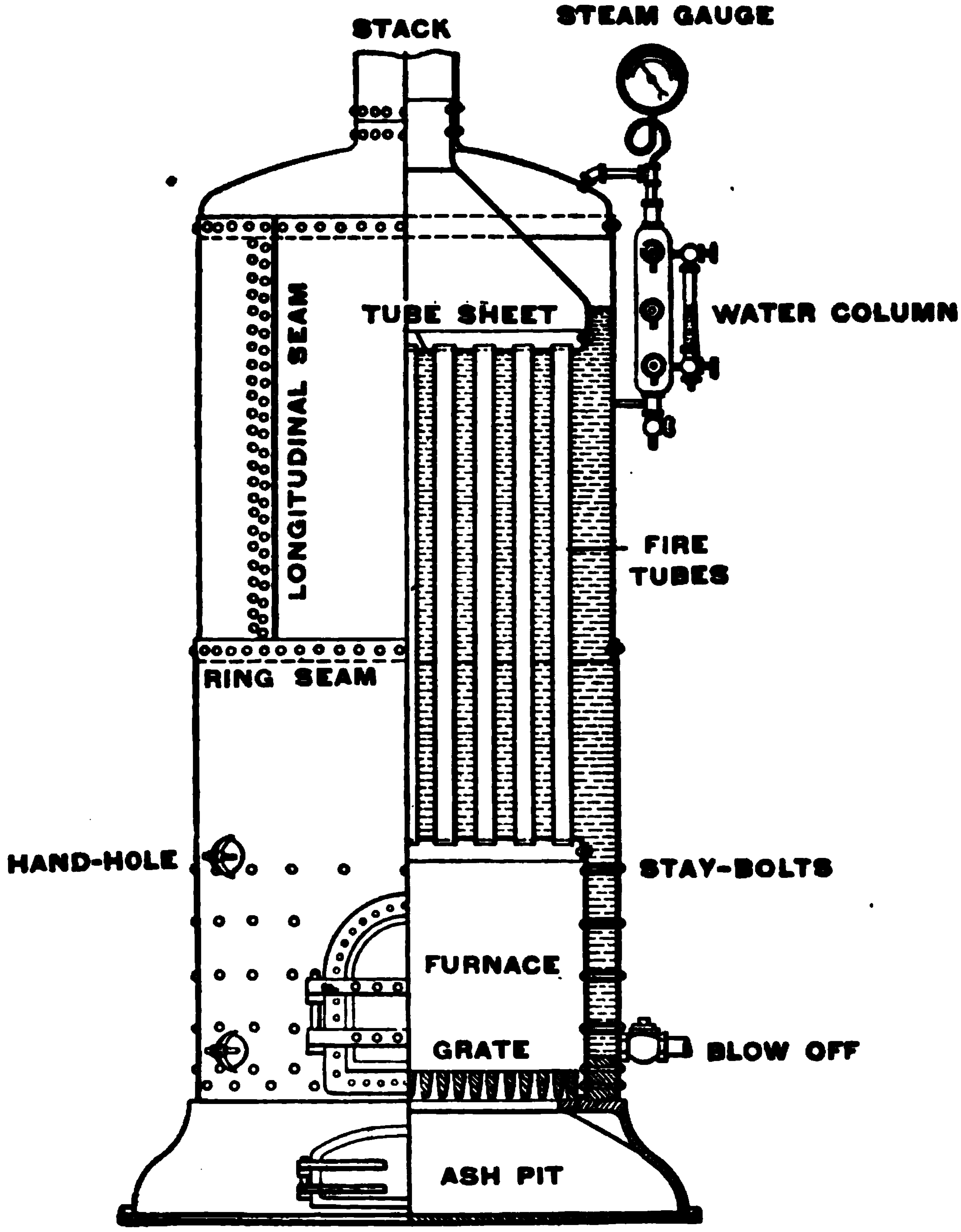


FIG. 23. Vertical Tubular Boiler with Submerged Tube Sheet.

from unequal expansion and liability to overheating is of sufficient moment to justify the use of the submerged type, particularly where the boiler is likely to be forced above its rated capacity. The advantages of this type of boiler are (1) compactness and portability; (2) requires no setting beyond a light foundation; (3) is a rapid steamer, and (4) is low in first cost. The disadvantages are (1) inaccessibility for thorough inspection and cleaning; (2) small steam space, which results in excessive

priming at heavy loads; (3) poor economy except at light loads, as the products of combustion escape at a high temperature on account of the

SECTIONAL FRONT ELEVATION

Fire-Tube Boiler.

shortness of the tubes; (4) smokeless combustion practically impossible with bituminous coals; (5) the small water capacity results in rapidly fluctuating steam pressures with varying demands for steam.

Although vertical fire-tube boilers are usually of very small size, being seldom constructed in sizes over 60 horse power, an exception is found in the Manning boiler, Fig. 24, which is constructed in sizes as large as 250 horse power. Many of the disadvantages found in the smaller types are obviated in the Manning boilers, which, as far as safety and efficiency are concerned, rank with any of the other first-class types. They differ from the boiler described above mainly in having the lower or furnace portion of much greater diameter than the upper part which encircles the tubes. This permits a proper proportion of grate, which is not obtainable in boilers like Figs. 1 and 23. The double flanged head connecting the upper and lower shells allows sufficient flexibility between the top and bottom tube sheets to provide for unequal expansion of tubes and shell. The ash pit is built of brick and the water leg does not extend below the grate level, thus doing away with dead water space. Where overhead room permits and ground space is expensive, this boiler offers the advantage of taking up a small floor space as compared with horizontal types.

61. Fire-Box Boilers. — Although vertical fire-tube boilers may be classed as fire-box boilers, yet the term "fire box" is usually associated with the locomotive types, whether used for traction or stationary purposes. The usual form of fire-box boiler as applied to stationary work

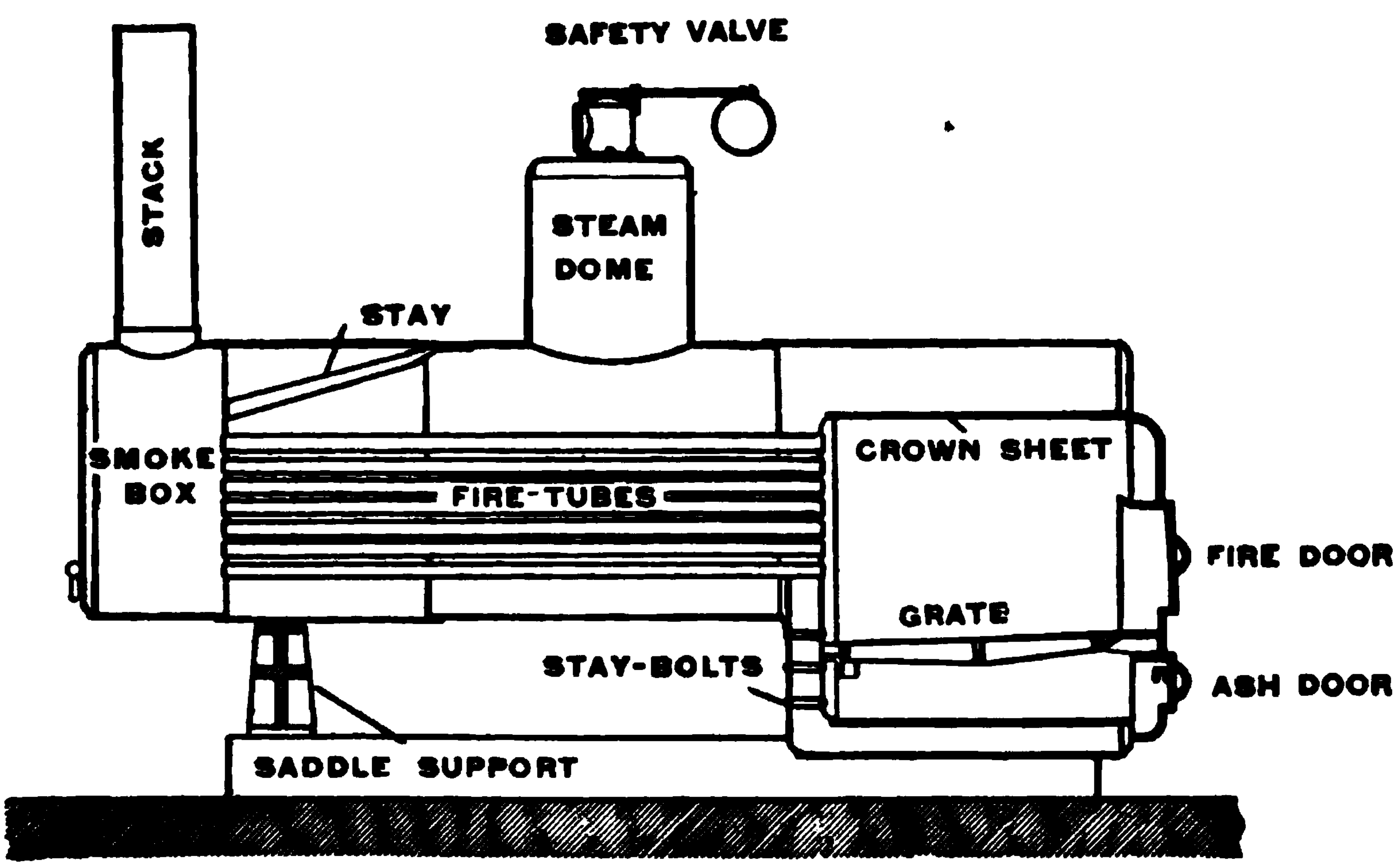


FIG. 25. Typical Fire-box Boiler. — Stationary Type.

is illustrated in Fig. 25. The shell is prolonged beyond the front tube sheet to form a smoke box. The front ends of the tubes lead into the smoke box and the rear ends into the furnace or fire box. The fire box

is ordinarily of rectangular cross section, and is secured against collapse by stay bolts and other forms of stays. In Fig. 25 the smoke box is of cylindrical cross section and hence requires no staying except at the flat surface. Fire-box boilers are used a great deal in small heating plants where space limitation precludes other types. Their steam capacity gives them an advantage over the vertical tubular form. Being internally fired no brick setting is required. They are usually of cheap construction, designed for low pressure, and seldom made in sizes over 75 horse power. Unless carefully designed and constructed high steam pressures are apt to cause leakage because of unequal expansion of boiler shell, tubes, and fire box. Portable fire-box boilers with return tubes are made in sizes as large as 150 horse power and for pressures as high as 150 pounds per square inch, but being more costly than some of the other types of boilers of equal capacity are used only where portability is an essential requirement.

62. Scotch-Marine Boiler. — Where an internally fired boiler is desired for large powers the Scotch-marine type is finding much favor with engineers. A number of the tall office buildings in Chicago are equipped with boilers of this class which are giving good results. They require little overhead room, no brick setting, and are excellent steamers.

FIG. 26. Stationary Scotch-Marine Boiler.

The Continental boiler, Fig. 26, is one of the best known of this type. The boiler is self-contained and requires no brick setting, the only fire brick used being those that form the bridge wall, baffle ring and the layer at the back of the combustion chamber. The furnace and tubes are

entirely surrounded by water, so that all fire surfaces, excepting the rear of the combustion chamber, are water cooled. The furnace is corrugated for its whole length. These corrugations, in addition to giving greater strength to the furnace, act as a series of expansion joints, taking up the strains due to unequal expansion of furnace and shell. Practically all types of mechanical stokers and grates are applicable to these boilers. The advantages of a Scotch boiler and of all internally fired boilers are (1) minimum radiation losses; (2) requires no setting; (3) no leakage of cool air into the furnace as sometimes occurs through cracks or porous brickwork of other types; (4) large steaming capacity for the space occupied. The circulation, however, is not always positive and the water below the furnace may be considerably below the average or normal temperature, giving rise to unequal expansion and contraction which may cause leakage. The boiler proper is relatively costly, but this is offset to some extent by the absence of setting.

63. Robb-Mumford Boiler. — Fig. 27 shows a section through a Robb-Mumford boiler, which is a modification of the Scotch-marine

FIG. 27. Robb-Mumford Boiler.

and of the horizontal tubular type. It consists of two cylindrical shells, the lower one containing a round furnace and tubes and the upper one forming the steam drum, the two being connected by two necks. The lower shell has an incline of about one inch per foot from the horizontal, for the purpose of promoting circulation and draft, and also for convenience in washing out the lower shell. Combustion takes place in the furnace, which is surrounded entirely by water, and

the gases pass through the tubes and return between the lower and upper shells (this space being enclosed by a steel casing) to the outlet at the front of the boiler. Mingled water and steam circulate rapidly up the rear neck into the steam drum, where the steam is released, the water passing along the upper drum towards the front of the boiler and down the front neck, a semicircular baffle plate around the furnace causing the down-flowing water to circulate to the lowest part of the lower shell under the furnace. The outer casing, which incloses the space between the lower and upper shells, including the rear smoke box and the smoke outlet, is constructed of steel plate, with angle-iron stiffeners, the various sections being bolted together for convenient removal. The inside of the steel case, including the rear smoke chamber, is lined with asbestos air-cell blocks fitted in between the angle-iron stiffeners. The top of the upper drum and bottom of the lower shell are also covered with non-conducting material after the boiler is erected. Owing to the fact that steam and water spaces are divided between two cylindrical shells, the thickness of plates is not so great as in the Scotch-marine or horizontal return tubular types; and the rear chamber of the marine boiler is avoided.

The chief claim for this type of boiler is compactness. A battery of five 200-horse-power units occupies a floor space of but 33 feet in width by 20 feet in depth and 12.5 feet high. Each unit is entirely independent and may be isolated for cleaning, inspection, and repairs.

64. Horizontal Return Tubular Boilers. — These are the most common in use and are constructed in sizes up to 200 horse power. They are simple and inexpensive and, when properly operated, durable and economical. Figs. 28 to 31 show various forms of standard settings, and Figs. 75, 76, and 76a different " smokeless " settings. The grate is independent of the boiler, and the products of combustion pass beneath the shell to the back end, returning through the tubes to the front, and into the smoke connection.

The tubes are 3 to 4 inches in diameter and from 14 to 18 feet long, and are expanded into the tube sheets. The portion of the tube sheets not supported by the tubes is secured against bulging by suitable stays. Access to the interior of the boiler is obtained through manholes. The most convenient arrangement for inspection and cleaning is to have one manhole located at the top of the shell and one at the bottom of the front tube sheet. Return tubular boilers are made either with an extended front (Fig. 28) or flush front (Fig. 29). The latter costs a little more for brick and setting, but it is more convenient to operate and the boiler is less expensive. The shell may be supported by lugs on the brickwork as in Fig. 28 or by steel beams and hangers as in Fig. 30.

SETTING FOR 72 INCH
RETURN TUBULAR BOILER
AS ADVOCATED BY THE
HARTFORD BOILER INSPECTION
AND INSURANCE CO.

SECTIONAL PLAN.

SECTIONAL FRONT ELEVATION

LONGITUDINAL SECTION

Fig. 28. Return Tubular Boiler Setting. — Extended Front.

SECTIONAL PLAN
THROUGH FIRE-DOORS

SETTING FOR STATIONARY
FULL-ARCH FRONT
BOILERS

Designed and Advocated by the
Erie City Iron Work

FRONT ELEVATION

FIG. 29. Return Tubular Boiler Setting. — Flush Front.

The latter construction permits the brickwork and shell to expand or contract independently, and settling of the brickwork does not affect the boiler alignment. With the side bracket support, the front lugs usually rest directly on iron or steel plates imbedded in the brickwork, and the back lugs on rollers, to permit free expansion and contraction. The brackets are long enough to rest upon the outside wall, so that the inside brick lining can be renewed without disturbing the setting. The distance between the rear tube sheet and wall should be about 16 inches

FIG. 30. Return Tubular Boiler Setting. — Steel Beam Suspension.

for boilers less than 60 inches in diameter and from 20 to 24 inches for larger ones. The distance between grate and boiler shell should not be less than 28 inches for anthracite coal and 36 inches for bituminous coal.* The greater this distance the more complete the combustion, since the gases will have a better opportunity for combining with the air before coming in contact with the comparatively cool surfaces of the shell. The shell should be slightly inclined toward the blow-off end so as to drain freely.

The vertical distance between the bridge wall and shell is usually between 10 and 12 inches. The lower part of the combustion chamber behind the bridge wall may be filled with earth and paved with common

* For smokeless combustion the setting must be modified. See furnace illustrated and described in paragraph 96.

FIG. 31. Boiler Setting; "Wood" Mill of the American Woolen Company, Lawrence, Mass.

brick, as in Fig. 31, or left empty as in Fig. 29. The shape of the walls, whether curved to conform to the shell or flat, appears to have little influence on the economy.

The side and end walls are ordinarily constructed of common brick with an inner lining of fire brick, and may be solid as in Fig. 29 or double with air spaces as in Fig. 28. The latter construction is preferable and permits the inner and outer walls to expand independently without cracking and settling. The side walls are braced by five pairs of buck-staves, with through rods under the paving and over the tops of the boilers.

The connection between the rear wall and the shell is a source of more or less trouble on account of the expansion and contraction of the boiler. Cast-iron supports of T section supporting a fire-brick arch are usually employed as illustrated in Fig. 32, the clearance between the arch and the shell being sufficient to allow the necessary expansion.

Fig. 32. Furnace Arch Bars.

Fig. 33. Back connection made with Cast-iron Plate.

Fig. 33 shows the common method of resting one end of the arch supports on the rear wall and the other end on an angle iron riveted to the boiler.

The products of combustion are sometimes carried over the top of the boiler as shown in Fig. 31. This tends to superheat the steam, but the advantage gained is probably offset considerably by the extra cost of the setting and the accumulation of soot on the top of the shell. The arrangement is not common.

The steam connection is naturally made to the highest point in the boiler shell. Frequently a steam dome, to which the steam nozzle is connected, is provided as in Fig. 29. The function of the steam dome is to increase the steam space so as to permit the collection of dry steam at a point high above the water level. If a boiler is too small for its work

LINE

STEEL SUPPORT

FIRE

ASH

BRIDGE WALL

FIG. 34. Babcock and Wilcox Boiler and Setting.

and is forced far above its rating a steam dome is probably an advantage, though its use is less common now than formerly, since a properly designed boiler insures ample steam space without one. A dry pipe inside the boiler above the water line as in Fig. 26 or 27 is commonly used to guard against priming where the nozzle is connected to the shell.

For low pressures and small powers the return tubular boiler has the advantage of affording a large heating surface in a small space and large overload capacity. It requires little overhead room and its first cost is low. On the other hand the interior is difficult of access for purposes of cleaning and inspection. Boilers of this type are seldom constructed in sizes above 150 horse power or for pressures over 150 pounds per square inch, since the cost increases rapidly as the pressure rises above this amount.

65. Babcock & Wilcox Boiler. — Fig. 34 shows a longitudinal section through a Babcock & Wilcox boiler, illustrating a typical horizontal water-tube type. The tubes, usually 4 inches in diameter and 18 feet

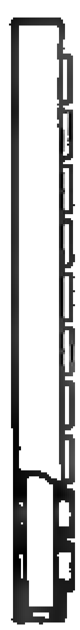


FIG. 35. Details of Header,—Babcock and Wilcox Boiler.

FIG. 36. Front Section, — Babcock and Wilcox Boiler.

in length, are arranged in vertical and horizontal rows and are expanded into pressed-steel headers. Two vertical rows are fitted to each header and are "staggered" as shown in Fig. 35. The headers are connected

with the steam drum by short tubes expanded into bored holes. Each tube is accessible for cleaning through openings closed by covers with ground joints held in place by wrought-iron clamps and bolts. The tubes are inclined at an angle of about 22 degrees with the horizontal. The rear headers are connected at the bottom to a cast-iron mud drum. The steam drum is horizontal and the headers are arranged either vertically or at right angles to the tubes. The boiler is supported by steel girders resting on suitable columns independent of the brick setting. The grate is placed under the higher ends of the tubes, the products of combustion passing at right angles to the tubes and being deflected back and forth by fire-tile baffles. The feed water enters the front of the steam drum as shown in Fig. 36. A rapid circulation is effected by the difference in density between the solid column of water in the rear header and the mixed steam and water in the front one. B. & W. boilers under 150 horse power have but one steam drum, and the larger sizes have two. The number of tubes varies with the size of boiler, ranging from 6 in width and 9 in height in the 100-horse-power boilers to 14 high and 18 wide in the 500-horse-power boilers.

66. Heine Boiler. — Fig. 37 shows a longitudinal section through a Heine horizontal water-tube boiler. This boiler differs from the B. & W. boiler in that the tubes are expanded into a single large header constructed of boiler steel. The drum and tubes are parallel with each other and inclined about 22 degrees with the horizontal. The feed water enters at the front of the steam drum and flows into the mud drum, from which it passes to the rear header. Steam is taken from the front of the steam drum and is partially freed from moisture by the dry pipe *A*. A baffle over the front header prevents an excess of water from being carried into the dry pipe. As the rear header forms one large chamber, no additional mud drum is necessary and the sediment is blown off from the bottom by the blow-off cock. The circulation is somewhat freer than in the B. & W. boiler on account of the large sectional area through the headers.

67. Wickes Boiler. — Fig. 38 shows a section through a Wickes vertical boiler, illustrating the vertical water-tube type. The steam drum and water drum are arranged one directly above the other. The tubes are expanded and rolled into both tube sheets and are divided into two sections by fire-brick tile. The water line in the steam drum is carried about two feet above the tube sheet, leaving a space of five feet between water line and top of the drum. This affords a large steam space and disengagement surface. Feed water is introduced into the steam drum below the water line and flows downward through the tubes of the second compartment. The boiler is supported by four

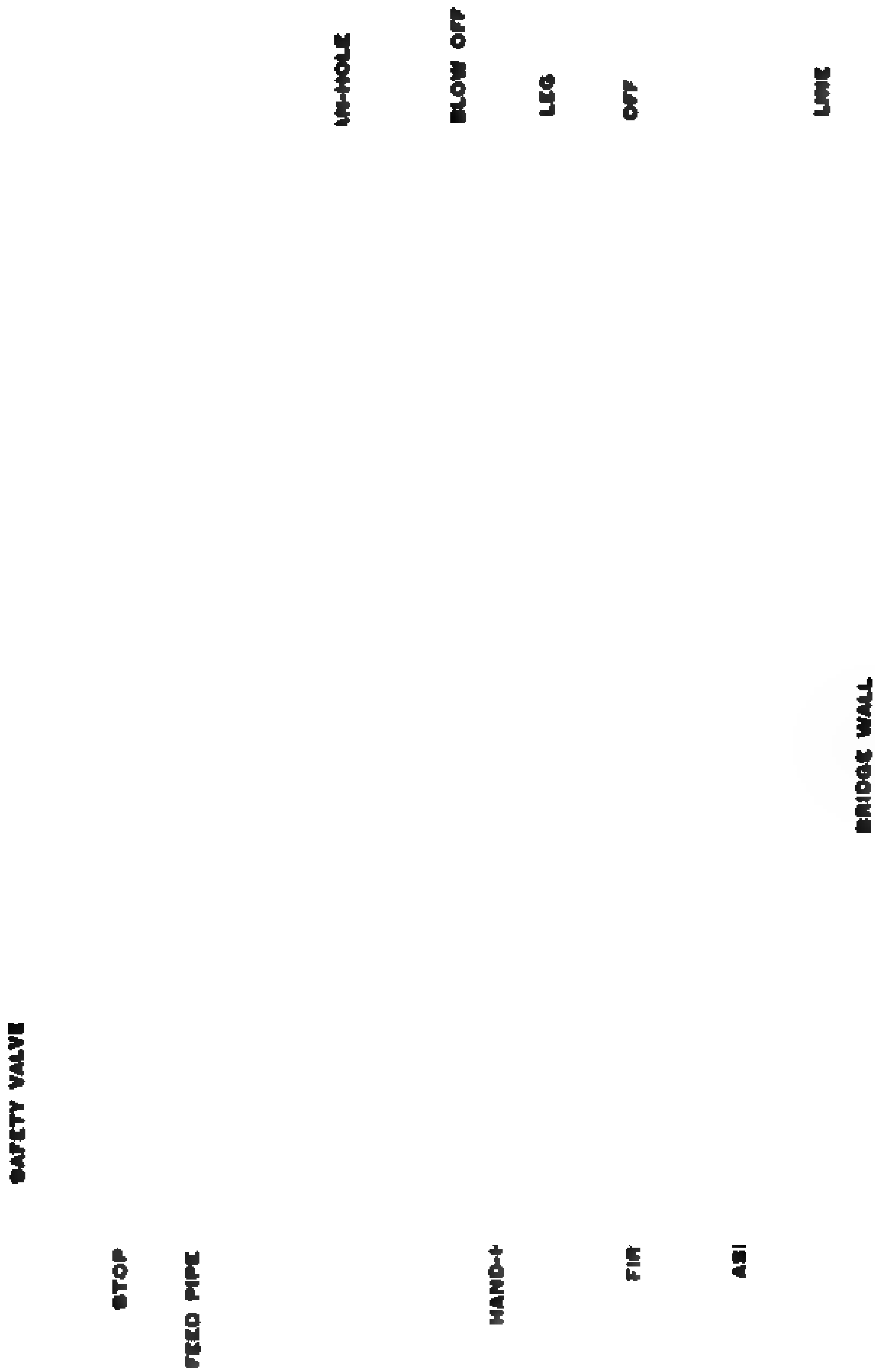


Fig. 37. Heine Boiler and Setting.

brackets riveted to the shell of the bottom drum and is independent of the setting. The entire boiler is enclosed in brickwork and is completely surrounded by the products of combustion. The upper part

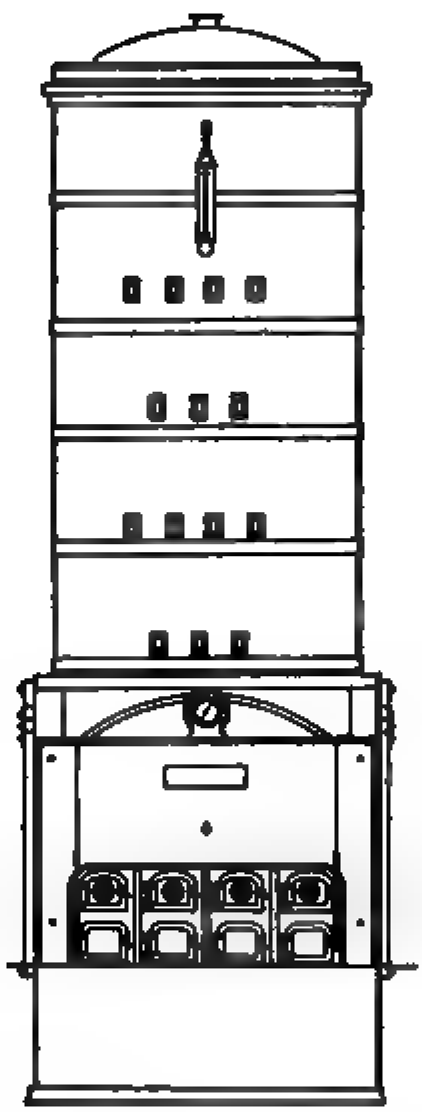


Fig. 38. Wickes Vertical Water Tube Boiler.

of the steam drum acts as a superheating surface and tends to dry the steam. Wickes boilers are simple in design, easy to inspect and clean, low in first cost, and comparable in efficiency with any water-tube type of boiler.

FIG. 33a. 1200-H P. Parker Down-Flow Boiler with Double-Ended Setting.

67a. Parker Boiler. — Fig. 38a shows a longitudinal sectional elevation and an end sectional elevation of a 1200-H.P. Parker Down-Flow Boiler with double-ended setting. This type of boiler is finding much favor with engineers for central stations where large units are desired. The Parker boiler differs from the conventional horizontal water-tube boiler principally in circulation and flexibility.

Feed water is pumped into the economizer or feed element (1), Fig. 38a, at *O, O*, and flows downward through a series of tubes, discharging finally into the drum through an upcast *H*. In a large unit, as illustrated here, there are two feed elements and two drums. The circulation in the feed element is indicated by solid lines and arrow points at the left of the end sectional elevation, the tubes having been omitted from the drawing for the sake of clearness.

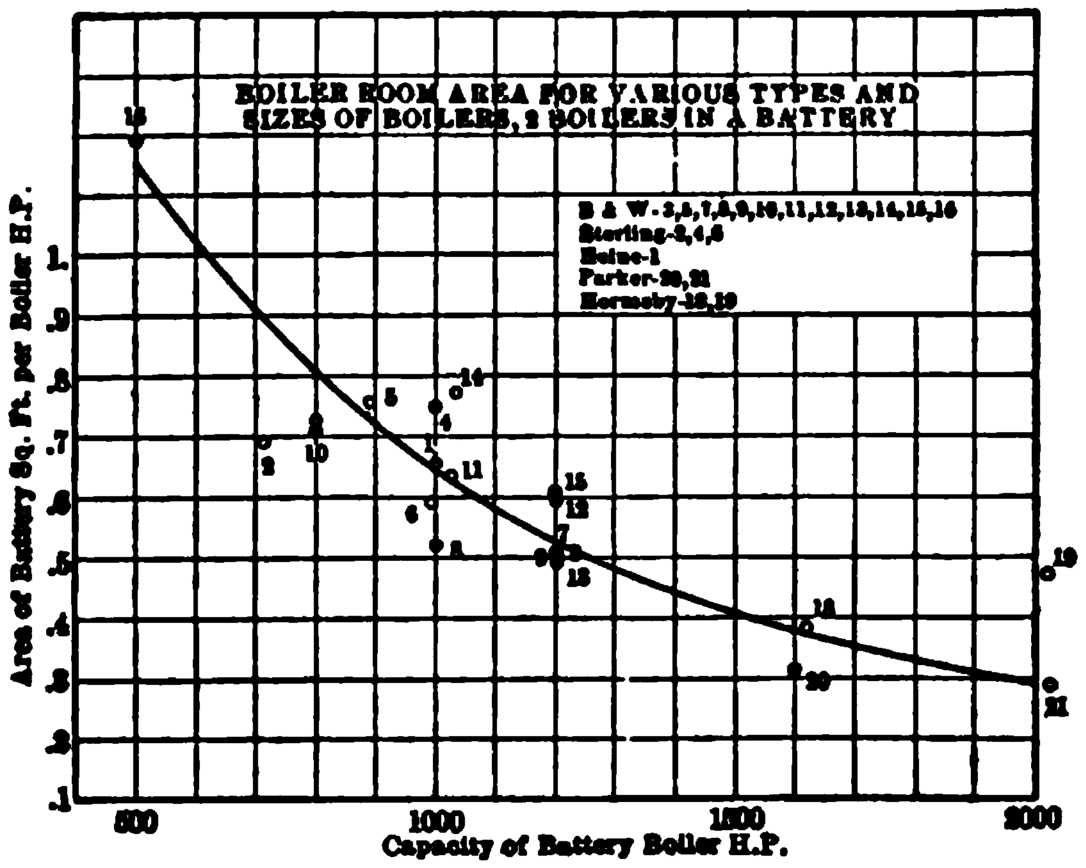


Fig. 38b.

The intermediate elements (2) take their water supply from the bottom of the drum through a cross-box *V*, the circulation being downward, as indicated by arrow points, through four tube wide elements, and finally discharge it through an upcast *X* into the steam space of the drum. Each element has a "down-comer" and an upcast. In the smaller sized boilers the intermediate elements are omitted.

The evaporator elements (3) take their water supply from the bottom of the drum at *V*, the circulation being downwards through two tube wide elements, and finally discharge it into the drum at *U*. The last two passes of the water are through the two bottom tubes of each element, thus assuring dry steam without the use of dry pipes. To prevent reversal of flow each element is fitted with a check valve at the admission end. Each drum is equipped with a diaphragm, as indicated, separating the steam and water spaces, thus insuring against foaming and priming.

Saturated steam is taken from the drum at *A* and passes by way of *B* to *C*, where it enters the superheater *S*. The superheated steam leaves the superheater at *D* and passes by way of *E* and *R* to the storage drum *N*, finally leaving the boiler at *G*. The superheater is designed

to maintain an approximately constant degree of superheat for all variations in load.

All tubes are connected by malleable-iron junction boxes, the interior of each tube being accessible through hand holes placed opposite the

FIG. 39. Stirling Boiler and Setting.

end of each tube. The hand-hole cover plates are on the inside of the box and have conical ground joints, thus dispensing with gaskets.

The Parker boiler is built single or double ended, with or without superheater, and in sizes ranging from 50 H.P to 2500 H.P. standard rating.

68. Stirling Boiler. — Fig. 39 shows a longitudinal section through a Stirling water-tube boiler, which differs considerably from the types just described. Three horizontal steam drums and one horizontal mud drum are connected by a series of inclined tubes. The tubes are bent at the ends to permit them to enter the drums radially. Short tubes

connect the steam spaces of all the upper drums and also the water spaces of the front and middle drums. Suitably disposed fire-tile baffles between the banks of tubes direct the gases in their proper course. The boiler is supported on a structural steel framework independent of the setting. The feed water enters the rear upper drum, which is the cooler part of the boiler, and flows to the bottom or mud drum, where it is heated to such an extent that many of the impurities are precipitated. There is a rapid circulation up the front bank of tubes to the front drum, across to the middle drum, and thence down the middle bank of tubes to the mud drum. The interior of the drums is accessible for cleaning by manholes located in the ends. The Stirling furnace is distinctive in design. A fire-brick arch is sprung over the grates immediately in front of the first bank of tubes. The large triangular space between boiler front, tubes, and mud drum forms the combustion chamber. Stirling boilers are somewhat lower in first cost than other types of water-tube boilers on account of the absence of numerous hand holes and the like which are necessary in the horizontal type.

69. Unit of Evaporation. — The performance of a boiler and furnace may be expressed in terms of the weight of water evaporated per hour per square foot of heating surface or of the weight evaporated per pound of fuel. To reduce all performances to an equal basis so as to facilitate comparison the evaporation under actual conditions is conveniently referred to the equivalent evaporation from a feed-water temperature of 212 degrees F. to steam at atmospheric pressure. The heat required to evaporate one pound of feed water at a temperature of 212 degrees F. into steam of the same temperature, or "from and at 212 degrees" as it is commonly called, is 965.7 B.T.U.* The ratio of the heat necessary to evaporate one pound of water under actual conditions of feed temperature and steam pressure to the heat required to evaporate one pound from and at 212 degrees is called the factor of evaporation. Thus for dry steam,

$$F = \frac{\lambda - t + 32}{965.7}, \tag{11}$$

in which

F = factor of evaporation.

λ = total heat of one pound of steam at observed pressure.

t = temperature of the feed water, degrees F.

* Recent redeterminations of the properties of saturated steam give this figure as 970.4.

If the steam is wet,

$$\lambda = xr + q, \tag{12}$$

in which

q = heat in liquid at observed pressure.
x = the quality of the steam.
r = latent heat of evaporation at observed pressure.

If the steam is superheated,

$$\lambda = r + q + Ct_s, \tag{13}$$

in which

C = the specific heat of the superheated steam.
t_s = the degree of superheat, degrees F.

69a. Heat Transmission. — Fig. 39a shows a section through a boiler heating plate and serves to illustrate the accepted theory of heat transmission. The outer surface of the plate is covered with a thin layer of soot and a film of gas, and the inner surface is similarly protected by a layer of scale and a film of steam and water. It is therefore reasonable to assume that the *dry* surface of the plate is located somewhere within the film of gas, and the *wet* surface within the film of water and steam.

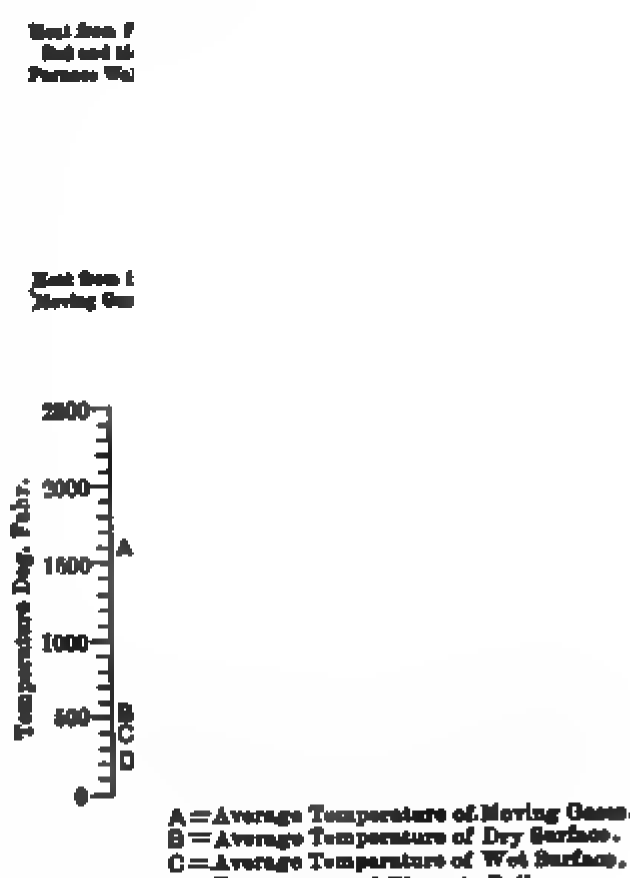


FIG. 39a Heat Transmission through Boiler Plate.

The heat is imparted to the *dry* surface by (1) *radiation* from the hot fuel bed and furnace walls, and by (2) *convection* from the moving furnace gases. The heat is transferred through the boiler plate and its coatings purely by *conduction*. The final transfer from the *wet* surface to the boiler is mainly by *convection*.

Radiation depends on the temperature, and according to the law of Stephen and Boltzmann is approximately proportional to the difference between the fourth power of the absolute temperature of the fuel bed and furnace walls and the temperature of the dry surface of the heating plate. According to this law the heat transmitted by radiation increases rapidly with the increase in furnace temperature. In the modern boiler the surface exposed to radiation is only a small portion of the total

heating surface, and, since in well-operated furnaces the temperature of the furnace cannot be increased materially on account of practical considerations, there is little hope of increasing the capacity of a boiler by increasing the furnace temperature.

The heat imparted to a boiler plate by *convection* may be determined by the following equation (Prof. Perry, "The Steam Engine," 1906 Ed., p. 588):

$$H = C\,(t_1 - t_2)\,vd, \tag{13a}$$

in which

H = B.T.U. transferred per hour per sq. ft. of heating surface.
C = a coefficient determined by experiment.
t_1 = temperature of the moving gases, degrees F.
t_2 = temperature of the dry plate surface, degrees F.
v = velocity of the gases, feet per sec.
d = density of the gases, lbs. per cubic foot.

Prof. Nicholson gives the following modifications of formula (13a) as applied to boiler tubes or flues (Engr. Lond., Feb. 19, 1908):

$$H = \left\{\frac{t}{200} + \frac{1}{40\,t}\left(1 + \frac{1}{m}\right)\right\}(t_1 - t_2)\,vd, \tag{13b}$$

in which

t = mean film temperature.
m = hydraulic radius = area of tube in square inches ÷ perimeter of the tube in inches, other notations as in (13a).

Both equations are based upon the same general law except that the latter gives a means of determining coefficient C in terms of the mean film temperature and the dimensions of the flues or tubes.

An examination of equation (13a) shows that for a given set of conditions the heat imparted to a unit of dry surface of heating plate varies directly as the difference between the temperature of the hot gases and that of the dry surface and directly as the velocity and density of the gases. However, the density of the gases drops with the rise of temperature, and increase in furnace temperature does not necessarily imply increase in heat impartation. It is the utilization of the *velocity* factor, then, which offers a possibility of increasing boiler capacity and efficiency.

Experiments by Prof. Nicholson and the U. S. Geological Survey show that by establishing a powerful scrubbing action between the gases and the boiler plate the protecting film of gas is torn off as rapidly as it is formed and new portions of the hot gases are brought into contact with the plate, thereby greatly increasing the rate of heat transmission. Similarly the faster the circulation of the water the greater will be the

scrubbing action tending to remove the bubbles of steam from the wet surface and the more rapid will be the transfer from the plate to the boiler water.

The resistance of the metal itself is so small that it may be neglected in calculating the heat transmission, and it may be logically assumed that the plate will take care of all the heat that reaches its dry surface.

Prof. Nicholson found that by filling up the flue of a Cornish boiler with an internal water vessel, leaving an annular space of only 1 inch around the latter, an evaporation eight times the ordinary rate was effected at a flow of gases 330 feet per second (8 to 10 times the average flow). The fan for creating the draft consumed about 4½% of the total power.

The conclusion is that the heating surface for a given evaporation at the present rating may be reduced as much as 90% for the same output, with a corresponding reduction in the size, cost and space requirements, or with a given heating surface of standard rating the output may be enormously increased; also the increase in power necessary to create the draft is by no means comparable with the advantages gained.

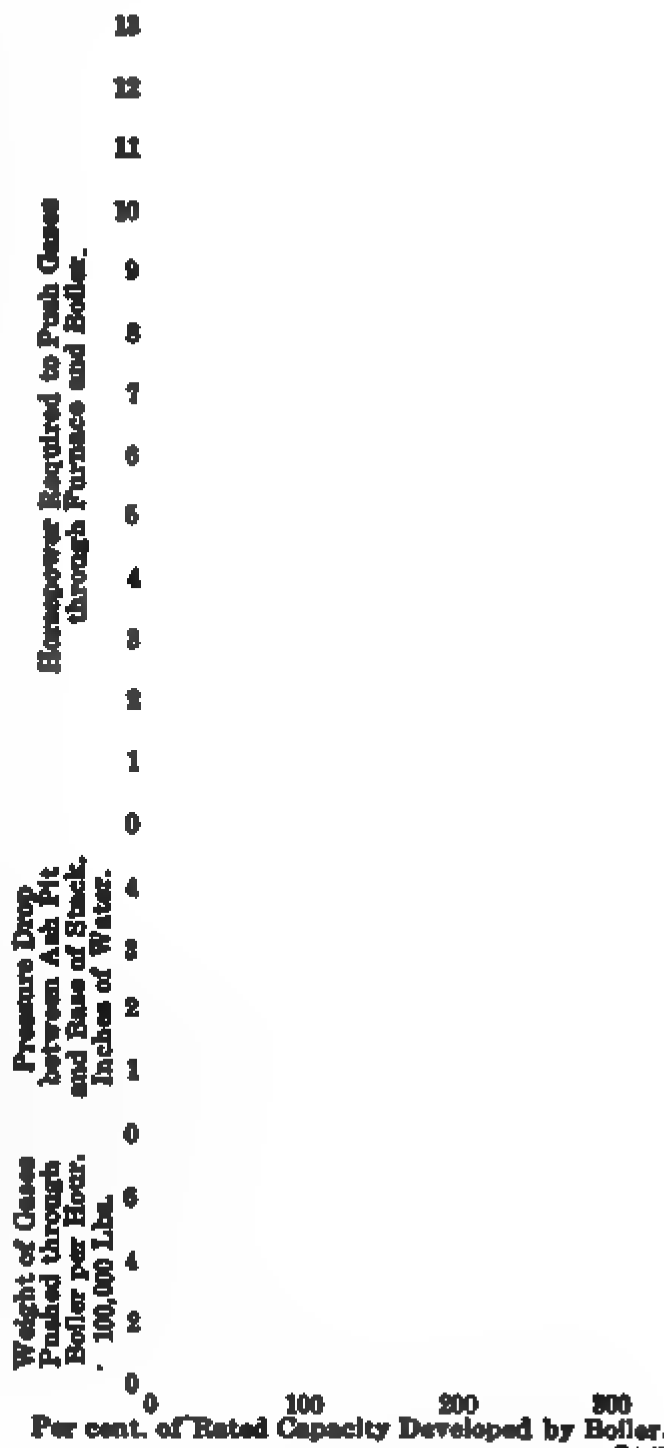


FIG. 39b. Influence of Draft on the Capacity of a Normand Water-Tube Boiler on the U. S. Torpedo Boat "Biddle."

The modern locomotive boiler is the nearest approach to these conditions in practice. Here a powerful draft forces the heated gases through small tubes at a very high velocity and an enormous evaporation is effected with a comparatively small heating surface. See Fig. 39b for influence of draft on the capacity of a torpedo boat boiler (Power and Engr., May 24, 1910).

These principles have been applied to a limited extent to stationary boilers already installed by making the gas passages smaller as compared to the length by means of suitable baffles (Fig. 38a) and by forcing larger weights of gas through the boiler, either by forced draft or by increasing the grate area (Fig. 68a).

In a general sense when the capacity of a boiler is doubled or tripled the over-all efficiency of the whole steam-generating apparatus drops, but the advantage gained usually offsets the loss in fuel economy. A close examination of the results, however, will show that the loss in efficiency is due more to low furnace efficiency than to inability of the boiler to absorb the heat generated.

In view of recent experiments it is not unlikely that within the next ten years boilers will be constructed capable of developing a boiler horse power with two or three square feet of heating surface instead of ten square feet, as at present, and with high over-all efficiency. (See Figs. 41a and 41b.)

Heat Transmission in Boilers, Kreisinger and Ray: Power and Engr., June 29, 1909, p. 1144; U. S. Geological Survey, Bulletin Journ. West Soc. Engrs., Sept. 18, 1907; Am. Inst. Elect. Engrs., Dec. 13, 1907.

Heat Transfer and Future Boiler Practice: A. H. Allen, Power and Engr., Sept. 21, 1909, p. 482; Engng., Lond, Feb. 19, 1908.

The Heat of Fuels and Furnace Efficiency: W. D. Ennis, Power and Engr., July 14, 1908, p. 50.

A Study in Heat Transmission (The Transmission of Heat to Water in Tubes as Affected by the Velocity of the Water), J. K. Clement and C. M. Garland, Univ. of Ill. Bulletin No. 40, Sept. 27, 1909.

70. Heating Surface. — All parts of the boiler shell, flues, or tubes which are covered by water and exposed to hot gases constitute the heating surface. Any surface having steam on one side and exposed to hot gases on the other is superheating surface. According to the recommendations of the American Society of Mechanical Engineers, the side next to the gases is to be used in measuring the extent of the heating surface. Thus measurements are made of the inside area of fire tubes and the outside area of water tubes. The heating surface in a boiler under average conditions of good practice is most efficient when the heated gases leave the uptake at a temperature of 100 to 200 degrees F. above that of the steam. Each square foot of heating surface is capable of transmitting a certain amount of heat, depending upon the *conductivity of the material, the character of the surface, the temperature difference between the gas and the water, the location and arrangement of the tubes, the density of the gas, the velocity of the gas, and the time allowed for transmission of the heat.* It is customary to assume a uniform heat

transmission for the entire surface. Thus with *most* boilers it is found that the best results are obtained with an evaporation of from 3 to 3.5 pounds of water from and at 212 degrees F. per square foot of heating surface, which is equivalent to allowing 12 to 10 square feet per boiler horse power. By increasing the quantity of heat the evaporation may be increased, but at the expense of efficiency, since a smaller percentage of the heat is utilized. For example, an evaporation as high as 20 pounds per square foot per hour has been effected in torpedo-boat practice, and 12 pounds per square foot per hour is not unusual in locomotive work, but such performances are invariably obtained at the expense of economy. The selection of the proper proportion of heating surface to the evaporation required is evidently a very important matter. For maximum economy under *average* conditions of operation, practice allows a proportion of 1 square foot to every 3.5 pounds of water to be evaporated from and at 212 degrees F. Where economy must be sacrificed to capacity, as in locomotive practice, a much higher evaporation is allowed.

The maximum evaporation is limited by the amount of coal which can be burned upon the grate. It the draft is sufficient, a good boiler can develop a horse power upon 0.75 to 0.5 of the surface recommended. In the very latest large central stations the gas passages and grate surface are proportioned so that the boiler may be operated at 100% above standard rating with high over-all efficiency.

The following table shows approximately the result which may be expected with different rates of evaporation.

POUNDS WATER EVAPORATED FROM AND AT 212 DEGREES F. PER SQUARE FOOT OF HEATING SURFACE PER HOUR.

2	2.5	3	3.5	4	5	6	8	10	12
PROBABLE RELATIVE ECONOMY.									
100	100	100	100	99	98	95	90	85	80

Efficiency of Boiler Heating Surface: Trans. A.S.M.E., 18–328, 19–571. Kent, *Steam Boiler Economy* (John Wiley & Son), Chapter IX. The *Nature of True Boiler Efficiency:* Jour. West. Soc. Engrs., Sept. 18, 1907. *Heat Transference through Heating Surface:* Engineering, 77-1.

71. The Horse Power of a Boiler. — A boiler horse power is equivalent to the evaporation of 34.5 pounds of water per hour from a temperature of 212 degrees F. to steam at atmospheric pressure. This corresponds

to 33,305 B.T.U. per hour.* Since the power from steam is developed in the engine and the boiler itself does no work, the above measure of capacity is merely conventional. Thus one boiler horse power will furnish sufficient steam to develop about three actual horse power in the best compound condensing engine, but only one-half horse power in a small non-condensing engine. Boilers should be purchased on the basis of heating surface and not on the horse power rating, since one bidder may offer a boiler with say 5 square feet of heating surface per horse power and another with 10 square feet, both being capable of the required evaporation, but the one with a small heating surface (which will, of course, be the cheaper boiler) will do so only at an increased cost of fuel. Manufacturers ordinarily rate their boilers on the basis of 10 to 12 square feet of heating surface per horse power, and the power assigned is called the *builder's rating*. As this practice is not uniform, bids and contracts should always specify the amount of heating surface to be furnished. According to the recommendations of the American Society of Mechanical Engineers, " A boiler rated at any stated capacity should develop that capacity when using the best coal ordinarily sold in the market where the boiler is located, when fired by an ordinary fireman, without forcing the fires, while exhibiting good economy. And further, the boiler should develop at least one-third more than stated capacity when using the same fuel and operated by the same fireman, the full draft being employed and the fires being crowded; the available draft at the damper, unless otherwise understood, being not less than one-half inch water column.

In determining the boiler horse power required for a given engine horse power it is convenient to estimate the steam consumption of the engine under actual conditions and then ascertain the equivalent evaporation from and at 212 degrees F. For example, assume a single non-condensing engine developing 20 horse power to use 50 pounds of steam per horse power hour, or 1000 pounds steam per hour; steam pressure, 80 pounds per square inch; feed-water temperature, 120 degrees F. Required the boiler horse power necessary to furnish this quantity of steam.

From equation (11), the factor of evaporation is

$$F = \frac{\lambda - t + 32}{970.4} = \frac{1185.3 - 120 + 32}{970.4} = 1.131.$$

One thousand pounds of steam under the given conditions are there-

* With the new value of $r = 970.4$ in place of 965.7 this figure becomes 33,478.8.

fore equivalent to 1000 × 1.131 = 1131 pounds from and at 212 degrees F.

The boiler horse power necessary to furnish steam for the 20-horse-power engine will be

$$\text{Boiler horse power} = \frac{1131}{34.5} = 32.8.$$

Example: A 15,000 kilowatt steam turbine and auxiliaries require 14.7 pounds of steam per kilowatt-hour at rated load; steam pressure 200 pounds per square inch gauge; superheat 150 degrees F.; feed-water temperature, 179 degrees F.

Required the boiler horse power necessary to furnish this quantity of steam.

The heat furnished to the turbine and auxiliaries per kilowatt-hour is

$$w \left\{ \lambda + C_p t_s - (t - 32) \right\} = 14.7 \left\{ 1199.2 + 0.57 \times 150 - (179 - 32) \right\}$$

$$= 16{,}724 \text{ B.T.U.}$$

$$\text{Boiler horse power} = \frac{15{,}000 \times 16{,}724}{33{,}479} = 7500 \text{ (approx.)}.$$

Table 12 gives the required hourly evaporation per boiler horse power at various feed temperatures and steam pressures.

The following table shows approximately the relation between boiler horse power and heating surface for different ratios of evaporation:

EVAPORATION FROM AND AT 212 DEGREES F. PER SQUARE FOOT PER HOUR.

2	2.5	3.0	3.5	4	5	6	7	8	9	10
SQUARE FEET HEATING SURFACE REQUIRED PER HORSE POWER.										
17.3	13.8	11.5	9.8	8.6	6.8	5.8	4.9	4.3	3.8	3.5

Builders of return tubular and vertical fire-tube boilers allow 11 to 12 square feet of heating surface per horse power; water-tube boilers are rated at 10 square feet per horse power, and Scotch-marine boilers at 8 square feet per horse power.

72. Grate Surface.—The amount of fuel which can be burned per hour limits the amount of water evaporated per unit of time and depends

TABLE 12.

REQUIRED HOURLY EVAPORATION PER BOILER HORSE POWER AT VARIOUS FEED TEMPERATURES AND STEAM PRESSURES.

Feed Temperature Fahr.	Steam Pressure in Pounds by Gauge.																				
	0	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200
50	29.51	29.29	29.14	29.02	28.92	28.84	28.77	28.70	28.64	28.59	28.54	28.49	28.45	28.41	28.37	28.33	28.29	28.26	28.23	28.20	28.17
60	29.77	29.55	29.40	29.28	29.18	29.09	29.02	28.95	28.89	28.84	28.79	28.74	28.69	28.65	28.61	28.57	28.54	28.51	28.48	28.45	28.42
70	30.04	29.81	29.66	29.54	29.44	29.35	29.27	29.21	29.15	29.09	29.04	28.99	28.94	28.90	28.86	28.82	28.78	28.75	28.72	28.69	28.66
80	30.31	30.08	29.93	29.80	29.70	29.61	29.53	29.46	29.40	29.34	29.29	29.24	29.19	29.15	29.11	29.07	29.03	29.00	28.97	28.94	28.91
90	30.59	30.36	30.20	30.07	29.97	29.88	29.80	29.73	29.67	29.61	29.55	29.50	29.45	29.41	29.37	29.33	29.29	29.25	29.22	29.19	29.16
100	30.88	30.64	30.47	30.34	30.24	30.15	30.07	30.00	29.93	29.87	29.82	29.77	29.72	29.67	29.63	29.59	29.55	29.51	29.48	29.45	29.42
110	31.17	30.93	30.76	30.63	30.52	30.43	30.34	30.27	30.20	30.14	30.09	30.04	29.99	29.94	29.90	29.86	29.82	29.78	29.74	29.71	29.68
120	31.46	31.22	31.05	30.91	30.80	30.71	30.63	30.55	30.48	30.42	30.36	30.31	30.26	30.21	30.17	30.13	30.09	30.05	30.01	29.98	29.95
130	31.76	31.52	31.34	31.20	31.09	30.99	30.91	30.83	30.76	30.70	30.65	30.59	30.54	30.49	30.45	30.41	30.37	30.33	30.29	30.25	30.22
140	32.07	31.82	31.64	31.50	31.38	31.29	31.20	31.12	31.05	30.99	30.93	30.88	30.83	30.78	30.73	30.69	30.65	30.61	30.57	30.53	30.50
150	32.39	32.12	31.94	31.80	31.68	31.58	31.50	31.42	31.35	31.28	31.22	31.17	31.12	31.07	31.02	30.97	30.93	30.89	30.85	30.81	30.78
160	32.71	32.44	32.26	32.11	31.99	31.89	31.80	31.72	31.65	31.58	31.52	31.46	31.41	31.36	31.31	31.27	31.23	31.19	31.15	31.11	31.08
170	33.03	32.76	32.58	32.43	32.31	32.20	32.11	32.03	31.96	31.89	31.83	31.77	31.71	31.66	31.61	31.56	31.52	31.48	31.44	31.40	31.37
180	33.37	33.09	32.90	32.75	32.63	32.52	32 43	32.34	32.27	32.20	32.14	32.08	32.02	31.97	31.92	31.87	31.83	31.79	31.75	31.71	31.67
190	33.71	33.43	33.23	33.08	32.95	32.84	32.75	32.66	32.59	32.52	32.45	32.39	32.33	32.28	32.23	32.18	32.14	32.10	32.06	32.02	31.98
200	34.06	33.77	33.57	33.41	33.28	33.17	33.08	32.99	32.91	32.84	32.77	32.71	32.65	32.60	32.55	32.50	32.45	32.41	32.37	32.33	32.29
212	34.49	34.18	33.98	32.80	33.69	33.58	33.48	33.39	33.31	33.24	33.17	33.11	33.05	33.99	32.94	32.89	32.84	32.80	32.76	32.72	32.68

upon the extent and nature of the grate surface, the character of the fuel, and the draft. A liberal allowance of grate surface is usually desirable, particularly when the boilers aré to be forced, since too small a grate increases the labor of handling and cleaning fires and results in poor economy.

With good coal low in ash approximately equal results may be obtained with large grate surface and light draft and with small grate and strong draft, the amount of coal burned per hour being the same in both cases. Bituminous coal low in ash gives best results with high rates of combustion, provided the ratio of grate surface to heating surface is properly proportioned. Coals high in ash require a comparatively large grate surface, particularly if the ash is easily fusible, tending to choke the grate. Where a strong draft is available a smaller grate may be used than with moderate draft, as a thicker bed of fuel can be carried. The relation between draft and rate of combustion for various sizes of coals is shown in Fig. 116, paragraph 127.

A number of boiler tests made by Barrus ("Boiler Tests") showed that the best economy with anthracite coal hand fired was obtained with an average ratio of grate surface to heating surface of 1 to 36 and at a rate of combustion of approximately 12 pounds of coal per square foot of grate surface per hour. In these tests a variation in grate and heating surface ratio of 1 to 36 up to 1 to 46 gave practically no difference in economy. With bituminous coal the tests showed that an average ratio of 1 to 45 gave the best results and at a rate of combustion of 24 pounds of coal per square foot of grate surface per hour.

Tests made by Christie (Trans. A.S.M.E., 19-330) gave an average combustion of 13 pounds of anthracite per square foot of grate per hour for maximum efficiency and 24 pounds of bituminous.

Current central station practice gives *normal* rates of combustion approximately as follows (lbs. per sq. ft. per hr.):

Anthracite	15–20	Eastern bituminous	20–24
Semi-bituminous	18–22	Western bituminous	30–35

Table 13 gives the relation between heating and grate surface in a number of recent boiler installations using different kinds of coal, and is illustrative of current practice.

In proportioning the grate surface for a proposed installation the principal factor considered is the character of the fuel, a study being made of the various fuels available, and the one selected which gives the highest evaporation per dollar. The latter data may usually be obtained from records of plants using the same grade of fuel and grates similar to those intended for the proposed plant.

TABLE 13.

RATIO OF HEATING SURFACE TO GRATE SURFACE IN RECENT BOILER INSTALLATIONS.

Nature of Plants.	No. of Plants.	Type of Boiler.	Type of Grate.	Height of Chimney.	Character of Fuel.	Ratio of Heating to Grate Surface.
Central stations	10	Hor water tube.	Chain.....	200 feet and over.	Ill. screenings, 15 to 20% ash.	65
Do........	8	...do......	Roney.....	...do......	Bituminous.	60
Do........	6	...do......	Murphy...	...do......	do.	60
Do.........	9	...do......	Miscel's...	...do......	Anthracite.	40
Manufacturing plants	20	Return tubular.	Hand fired	150–175	Anthracite.	35
Office buildings	6	...do......	Shaking grates.	Over 200..	Bituminous.	48
Central station*	1	Babcock & Wilcox.	Roney.....	Over 200..	Bituminous.	31

* Two stokers, one at front and one at rear of setting. (*Power*, Jan. 7, 1908, p. 25.)

73. Boiler and Furnace Efficiency. — The efficiency of the boiler, including the grate, is expressed by the ratio between the heat absorbed by the boiler per pound of dry coal *fired* and the calorific value of one pound of dry coal. The efficiency of the boiler alone is taken as the ratio between the heat absorbed per pound of combustible *burned on the grate* and the calorific value of one pound of combustible. The combustible burned on the grate is equal to the coal as fired minus moisture and the total refuse in the ash pit. The calculation of these efficiencies is illustrated by the following example:

ANALYSIS OF COAL.

	Per Cent.
Moisture	8
Ash	12
Combustible	80
	100

	Pounds.
Water evaporated from and at 212° F. per pound of coal as fired	8.281

	Per Cent.
Total refuse in ash pit	16
Percentage of ash in refuse	13
Combustible in ash	3

	B.T.U.
Heating value per pound of coal as fired	11,680
Heating value per pound of dry coal = 11,680 ÷ 0.92	12,696
Heating value per pound of combustible = 11,680 ÷ 0.80	14,600

$8.281 \div 0.92 = 9.001$ = equivalent evaporation per pound of dry coal.

$9.001 \times 965.7 = 8,692$ = heat absorbed per pound of dry coal.

Efficiency of boiler and grate $= \frac{8,692}{12,696} = 68.49$ per cent.

Combustible *burned* on grate $= 100 - (8 + 16) = 76$ per cent.

$8.281 \div 0.76 = 10.896$ = equivalent evaporation per pound of combustible burned on the grate.

$10.896 \times 965.7 = 10,522$ = heat absorbed per pound of combustible.

Efficiency of boiler $= \frac{10,522}{14,600} = 72.07$ per cent.

The efficiency of the grate alone might be expressed

$$\text{Efficiency of grate} = \frac{\text{Efficiency of boiler and grate}}{\text{Efficiency of boiler}},$$

which is equivalent to

$$\text{Efficiency of grate} = \frac{\text{Combustible actually burned}}{\text{Combustible fired}},$$

the numerator being the coal fired less moisture and the refuse from the ash pit, and the denominator the coal fired less moisture and the ash as determined from the proximate analysis.

The efficiency of combustion is sometimes expressed in terms of the difference in temperature between fuel bed and flue gas:

$$\text{Efficiency of furnace} = \frac{T_f - T_c}{T_f - T_a}, \tag{14}$$

in which

T_f = temperature of the furnace.
T_c = temperature of the flue gas.
T_a = temperature of the air.

The efficiency of the furnace or combustion may also be stated (R. S. Hale, Trans. A.S.M.E., 20–769):

$$\text{Efficiency of furnace} = \frac{S + F}{H}, \tag{15}$$

in which

S = B.T.U. absorbed by the boiler per pound of dry coal.
F = B.T.U. lost in flue gases per pound of dry coal.
H = Calorific value of 1 pound of dry coal.

The heat *absorbed* by the boiler expressed in percentage of the heat available has been given the name *true boiler efficiency* by the U. S. Geological Survey and may be expressed

$$\text{True boiler efficiency} = \frac{T_f - T_c}{T_f - T_s}, \tag{15a}$$

in which

T_s = temperature of the steam (saturated); other notation as in (14).

74. Boiler Performances. — Table 14 is compiled from a number of tests of different types of boilers with various types of grates and

TABLE 14.

EXAMPLES OF CURRENT STEAM BOILER PERFORMANCES.

Reference No.	Type of Boiler.	Rated Horse Power.	Percentage of Rated Horse Power.	Type of Grate.	Character of Fuel.	Grate Surface (Square Feet).	Steam Pressure (Absolute).	Feed Water (Degrees F.).	Pounds Coal per Square Foot Grate per Hour.	Actual Evaporation per Pound of Coal.	Equivalent Evaporation per Pound of Dry Coal.	Equivalent Evaporation per Pound of Combustible.	Efficiency of Boiler and Furnace (per Cent).
1	Babcock & Wilcox	107	99	Hand fired	George's Creek, mine run	25	148.6	166	14.8	8.76	9.52	......	64.5
2	Do	107	130	....do	Pocahontas lump	25	111	151.4	20.1	10.48	11.50	12.38	73.8
1	Do	107	330	....do	Hacker's Run, lump	25	113	53.2	29.1	6.90	8.30	......	58.7
3	Do	150	76	Chain grate	Bituminous pea, Illinois	28	133	73.5	19.9	......	7.11	8.93	62.2
3	Do	150	123	....do	....do	28	128	63.2	34.9	......	6.53	7.95	56.7
4	Do	200	120	Hand fired	Big Muddy	50.5	138	104.2	20.8	......	......	9.37	67.0
5	Do	250	86	....do	Anthracite pea	56	134.1	212.5	9.8	10.4	10.9	13.2	......
6	Do	412	105	Roney stoker	New River	74.8	163.5	74.3	18	9.36	11.1	11.7	72.2
6	Do	412	128	....do	....do	74.8	158.5	71.3	21	9.68	11.57	12.4	74.9
6	Do	412	134	....do	Soldiers' Mine, run	74.8	154.4	64.3	22.4	8.63	10.38	11.35	69.9
7	Do	420	140	Bennis stoker	......	50	176.4	66.5	35.2	9.95	11.96	......	82.2
8	Do	500	102	Chain grate	Illinois screenings	77	195	167	32.1	......	7.3	10.47	69.0
9	Heine	210	.....	Hand fired	Kansas	40.5	106.4	192	18.1	7.09	......	......	65.6
9	Do	210	.....	....do	Alabama	40.5	101.1	178	21.5	6.92	......	......	65.7
9	Do	210	.....	....do	Arkansas	40.5	104.2	188.5	15.7	8.29	......	......	65.9
9	Do	210	.....	....do	West Virginia	40.5	96.7	188	18.1	7.96	......	......	68.3
3	Do	350	109	Chain grate	Bituminous screenings	72	126.9	63.5	23.9	6.4	7.66	9.64	66.8
10	Do	500	60	Hand fired	Coke breeze	80.5	......	......	20.5	4.5	4.78	......	......
3	Stirling	60	85	....do	Illinois lump	13.5	97	......	19.9	......	6.57	7.63	54.5
11	Do	200	115	......	Youghiogheny lump	52	144	168	13.3	......	11.44	12.1	79.5
11	Do	250	85	......	George's Creek	57	123.3	210	11.7	10.54	11.02	12.06	77.0
11	Do	300	106	......	Cumberland	48.9	136.1	116.6	21.7	9.08	10.38	11.79	76.7
12	Do	350	384	Chain grate	Carterville washed, No. 4	82.5	118	195	21.7	6.95	7.91	8.69	59.3
13	Do	517	90	American underfeed	Soldiers' Mine, run	......	162.5	64.5	......	8.82	10.62	11.53	70.1

13	Do	517	96	....do	New River	......	159.8	69.6	......	9.92	11.89	12.48	75.9
13	Do	517	136	....do	....do	......	154.4	71.2	......	9.11	10.88	11.57	69.4
3	Do	520	73	Hand fired	Illinois duff	50.9	125.6	178.5	21	......	6.24	8.11	53.8
3	Do	520	96	....do	Illinois lump	50.9	127.3	182.8	25.4	......	6.63	8.07	51.0
14	Do	620	220	Bennis stoker	......	72	196.8	96.2	71.9	8.84	10.4	......	84.9
15	Climax vertical	500	......	Hand fired	Nixon navigation	64	191.7	80	17.6	9.1	......	......	79.3
15	Do	1000	......	Wilkinson stoker	George's Creek	113.6	·96	37.3	27	8.96	......	......	72.7
16	Wickes horizontal	150	98	Shaking grate	Ohio bituminous	27	114	38.6	21.3	7.25	8.86	9.79	67.5
17	Do	200	98	Hawley down draft	Pocahontas, mine run	80.5	69	124.7	15.3	10.2	11.53	12.5	77.9
18	Scotch-marine	225	......	....do	....do	35	150.6	185	26	9.53	......	......	74.2
18	Do	225	......	....do	Maryland smokeless	35	151.2	162.4	30.7	......	......	......	79
3	Return tubular	40	130	Hand fired	Illinois lump	18.7	97	47.5	13.5	......	7.05	7.78	59.6
19	Do	54	110	....do	George's Creek	20	84	119	10.1	9.08	10.25	11.52	......
20	Do	71	58	....do	Anthracite screenings	25	70	89	7.1	7	8.17	9.66	......
3	Do	100	117	....do	Illinois lump	30	80	48.4	18.2	......	7.38	8.30	63.4
3	Do	100	105	....do	......	33	125.6	......	13.1	6.95	8.41	10.69	64
21	Do	150	53	....do	Anthracite chestnut, No. 2	30	95	44	10.3	7.52	9.10	10.78	......
22	Do	388	90	....do	Lackawanna broken	120	99.6	198.4	11.1	8.50	8.96	10.73	......
23	Do	400	152	....do	George's Creek	63	......	......	32.6	9.04	......	11.53	73.1
23	Do	400	185	Chain grate	....do	63	......	......	36.8	9.41	......	12.06	76.1
24	Vertical tubular	75	86	Hand fired	Anthracite broken	15.9	76	125	15.4	7.98	8.95	10.13	......
25	Do	150	70	....do	Pocahontas	28.3	99.5	73	13.6	9.99	11.69	12.7	80.2
26	Do	150	126	....do	....do	28.7	137	38.1	23	......	10.03	12.0	80
26	Do	250	136	Roney stoker	George's Creek	52	123	39	20.8	......	10.81	11.98	......
26	Do	250	225	....do	....do	52	122	39.6	38.3	......	9.72	10.48	......

1. Eng. Mag., April, 1904, p. 96. 2. Eng. Rec., July 25, 1903, p. 102. 3. University of Illinois Bulletin, Vol. 3, No. 22. 4. Eng. Rec., Feb. 19, 1898, p. 258. 5. Eng. Rec., March 9, 1901, p. 220. 6. Power, Dec., 1901, p. 26. 7. Eng., Lond., March 21, 1902, p. 286. 8. B. & W. "Steam," p. 121. 9. Coal-Testing Plant, U.S. Geological Survey. 10. Eng. Rec., May 7, 1904, p. 587. 11. Stirling Boiler Catalogue. 12. Test at Armour Inst. of Tech., 1907. 13. Power, Dec., 1901, p. 26. 14. Eng. Rec., April 8, 1905, p. 404. 15. Morrin Climax Catalogue. 16. Test by Professor Wagner, March 18, 1892. 17. Test by G. H. Barrus, June 16, 1897. 18. Eng. News, Feb. 24, 1898, p. 122. 19. Barrus' Boiler Trials, p. 79. 20. Barrus' Boiler Trials, p. 76. 21. Barrus, p. 129. 22. Barrus, p. 74. 23. Power, July, 1902, p. 16. 24. Barrus, p. 190. 25. Eng. Rec., April 2, 1898, p. 390. 26. Trans. A.S.M.E., 26–92.

TABLE 14a.

PRINCIPAL DATA AND RESULTS OF TESTS ON BOILER NO. 6, UNIT NO. 10, FISK ST. STATION. COMMONWEALTH EDISON CO., CHICAGO.

(B. & W. Boiler, "Standard" Setting.)

Water-heating Surface, 5000 Sq. Ft. Superheating Surface, 914 Sq. Ft. Chain Grate Surface, 90 Sq. Ft.

Test No.	Date, 1908.	Horse Power.	Eff'y, Per Cent.	H.P. per Sq. Ft. Grate.	Heat Lost in Refuse, Per Cent.	Total Heating Surface per H.P.	Super-heat of Steam, Deg. F.	Dry Coal per Sq. Ft. G. S. per Hour.
2	Mar. 9	873	67.4	9.70	2.8	6.76	197	41.2
4	" 10	873	69.0	9.52	2.8	6.89	195	39.1
6	" 11	852	67.3	9.47	2.8	6.93	189	38.9
8	" 16	836	65.3	9.29	6.4	7.06	174	39.5
10	" 17	870	68.8	9.67	5.0	6.78	180	39.3
14	" 19	920	66.2	10.22	9.2	6.42	187	43.7
16	" 23	900	69.5	10.00	4.0	6.56	181	40.5
18	" 24	916	69.1	10.18	5.5	6.44	190	41.6
20	" 26	912	69.2	10.13	4.4	6.48	179	41.2
22	" 27	906	67.7	10.07	4.1	6.52	194	42.5
24	" 30	925	69.8	10.28	2.8	6.38	179	41.6
26	" 31	894	69.4	9.93	5.2	6.60	170	40.6
28	Apr. 1	922	71.2	10.24	3.6	6.40	169	40.4
30	" 2	923	71.5	10.26	4.6	6.40	173	40.5
32	" 7	914	70.0	10.20	4.5	6.46	175	40.9
34	" 8	939	73.8	10.4	3.8	6.28	181	40.4
36	" 10	911	70.9	10.1	3.0	6.48	185	40.2
38	" 11	967	70.1	10.7	3.0	6.11	192	42.6
40	" 13	995	67.8	11.1	3.4	5.93	211	43.6
42	" 14	887	66.8	9.9	4.5	6.65	202	40.8
44	" 27	880	69.5	9.8	5.5	6.72	169	39.7
48	" 29	927	71.5	10.3	3.3	6.37	171	40.8
50	" 30	899	70.3	10.0	4.2	6.57	171	39.6
52	May 6	886	69.4	9.8	5.3	6.67	171	38.2
54	" 7	900	69.1	10.0	4.8	6.56	171	39.2
56	" 8	967	71.9	10.7	4.8	6.10	164	40.1
58	" 11	902	70.5	10.0	3.3	6.55	163	39.6
60	" 13	875	70.7	9.7	3.8	6.74	147	38.3
64	" 14	1102	72.0	12.2	4.8	5.35	180	43.2

TABLE 14a.

PRINCIPAL DATA AND RESULTS OF TESTS ON BOILER NO. 6, UNIT NO. 10, FISK ST. STATION. COMMONWEALTH EDISON CO., CHICAGO.

(B. & W. Boiler, "Standard" Setting.)

Water-heating Surface, 5000 Sq. Ft. Superheating Surface, 914 Sq. Ft. Chain Grate Surface, 90 Sq. Ft.

Draft		B.T.U. per Pound Dry Coal.	Ash in Dry Coal, Per Cent.	Ash in Refuse, Per Cent.	Uptake Temp. Deg. F.	CO_2, Per Cent.	Heat Lost up Stack (Dry Gas), Per Cent.
Over Fire.	In Uptake.						
.87	1.34	11,634	18.46	82.33	466	6.9	....
.78	1.25	11,759	16.81	81.36	461	6.7	....
.83	1.25	12,039	16.08	80.03	463	7.7	15.6
.94	1.34	11,993	15.91	67.42	477	7.6	16.8
.84	1.24	11,909	15.71	71.32	475	7.9	16.2
.99	1.41	11,768	16.04	63.78	479	8.5	15.4
.77	1.17	11,846	16.68	79.04	483	9.1	14.0
.81	1.25	11,800	16.39	71.98	484	8.3	15.8
.77	1.21	11,846	15.51	78.53	486	9.0	14.5
.78	1.22	11,659	17.59	80.58	494	9.2	14.6
.68	1.28	11,800	16.22	82.97	487	8.8	15.1
.70	1.24	11,752	16.18	76.84	484	8.8	15.1
.62	1.21	11,862	15.38	82.99	480	9.2	14.1
.58	1.40	11,800	16.02	78.37	480	9.1	14.4
.73	1.24	11,815	16.84	77.84	494	9.0	14.7
.72	1.25	11,659	18.06	82.27	504	8.9	15.3
.65	1.13	11,831	17.15	86.92	493	9.7	13.4
.70	1.24	12,002	16.05	84.39	502	9.0	15.1
.71	1.23	12,469	14.87	82.14	522	9.7	13.3
.63	1.09	12,049	15.17	78.12	500	9.5	13.3
.71	1.26	11,801	15.75	77.21	470	8.3	15.7
.68	1.23	11,769	18.59	84.04	472	8.7	14.2
.66	1.27	11,955	16.11	79.30	473	7.9	16.1
.62	1.20	12,360	13.63	74.59	476	8.8	14.5
.66	1.31	12,298	13.62	75.19	480	9.0	14.4
.66	1.29	12,423	13.37	75.61	474	9.4	13.3
.92	1.18	11,956	17.45	83.24	451	9.2	12.5
.76	0.98	11,971	17.45	80.99	443	10.0	11.2
.68	1.15	13,126	10.24	70.90	487	10.4	12.1

characters of fuel. Although some of the tests show a combined efficiency of boiler and grate as high as 85 per cent (Engr., Lond., March 21, 1902, p. 286), such a performance cannot be expected for continuous operation under the average conditions of practice. In pumping stations or in plants where there are no peak loads and the boiler may be operated under a practically constant set of conditions a continuous efficiency of 75 per cent has been realized with coal as fuel and 80 per cent with crude oil, though these figures are exceptional. In very large central stations, with the usual loads in the morning and evening, an average efficiency throughout the year of 65 per cent is possible, though a good figure is not far from 60 per cent. In large isolated stations with variable loads good practice gives an average of 60 per cent. Small stations though showing an efficiency as high as 75 per cent at times seldom average 50 per cent for the year. The usual discrepancy between efficiency as determined by special tests and everyday operation is due to the fact that the efficiency test is usually conducted under ideal conditions: the boiler surfaces are cleaned, the rate of combustion carefully adjusted for maximum economy, and special attention given to the firing, whereas in actual practice these refinements are seldom attempted. Much depends upon the efficiency of the boiler-room staff, the character of furnace and fuel, draft, and the load factor. From the commercial standpoint the performance is best expressed in terms of the "cost to evaporate 1000 pounds of water from and at 212," or the "pounds of water evaporated per $1 of coal." Table 15 gives the results of a number of tests, made at the Armour Glue Works, Chicago, Ill., showing the cost of evaporating water with different grades of Illinois coal. The results were obtained from hand-fired Stirling boilers.

Boiler Room Economies: Am. Elecn., Oct., 1901, p. 506, Sept., 1905, p. 472; Cassier's Mag., March, 1906, p. 373; Elec. World, March 4, 1905; Engr. U.S., May 1, 1905, p. 304, Jan. 1, 1907; Engr., June 4, 1907, p. 758; Eng. Rec., June 27, p. 685; Elecn., Lond., Aug. 5, 1904; Power, Aug., 1905, p. 484; Eng. Mag., Oct., 1901, March, 1903.

Care and Management of Boilers: Engr. U.S., March 1, 1902, p. 142, Feb. 15, 1904, July 15, 1904, Jan. 1, 1907; Engineering, Feb. 18, 1898, p. 211, July 15, 1898, p. 84; Eng. Mag., Feb., 1901, p. 877, Oct., 1901, p. 91, March, 1903, p. 896; Power, Sept., 1904, p. 467, May, 1905, p. 267, Dec., 1905, p. 742, Sept., 1906, p. 550; Am. Elecn., Feb., April, Sept., 1904; Mech. Eng., July 25, 1903; Engr., Lond., April 15, 1904; Elec. Rev., July 13, 1907.

75. Effect of Capacity on Efficiency. — In general, as the horse power of a boiler increases above normal capacity the over-all efficiency will decrease, due to the fact that the furnace and gas passages are *ordinarily* proportioned to effect an evaporation of about 3.5 pounds of water from and at 212 degrees F. per square foot of heating surface per hour at rated load, the temperature of the escaping gases being from

150 to 200 degrees above that of the steam. To increase the rate of evaporation more coal must be burned per unit of time and consequently a larger volume of gas is generated. The larger the volume of gas the higher will be its velocity, which *finally* reaches a point where heating surface is insufficient in extent to absorb the extra heat and as a consequence the flue gas escapes at a higher temperature, resulting in lower boiler and furnace efficiency. With properly proportioned grate, furnace and gas passages a boiler may be operated at 100% above standard rating with little or no decrease in over-all efficiency. Fig. 40 shows a case in which the efficiency decreased with the increase in capacity, and Fig. 41a illustrates increased efficiency for the higher rates of driving. These curves are of value simply as illustrations of the behavior in specific cases, and are not applicable to all types of boilers.

TABLE 15.

RESULTS OF COAL TESTS AT ARMOUR GLUE WORKS, CHICAGO, AUG. 17, 1905.

Date of Test.	Name and Kind of Coal.	Railroad Car Number.	Cost per Ton Delivered.	Cost to Evaporate 1000 Pounds of Water.	Pounds Water Evaporated per $1.00 of Coal.
March 5, 1905..	Williamson County Coal Co.'s, mine run	C. C. C. & St. L. No. 26368	$1.90	$0.1531	6,532
March 3, 1905..	Harden & Hafer, mine run	S. I. No. 5735	1.70	0.1231	8,123
June 14, 1905 ..	Crerar-Clinch & Co., 2" screenings.	I. C. No. 88362	1.50	0.1293	7,734
June 15, 1905 ..	....do...............	I. C. No. 88362	1.50	0.1218	8,210
June 16, 1905 ..	....do...............	I. C. No. 88362	1.50	0.1175	8,511
June 17, 1905 ..	Brackett Coal and Coke Co., lump.	C. & E. I., No. 8891.	1.65	0.122	8,197
June 19, 1905 ..	....do...............	C. & E. I. No. 5002	1.65	0.1212	8,251
June 20, 1905 ..	....do...............	C. & E. I. No. 5002	1.65	0.1352	7,396
July 1, 1905....	Kellyville Coal Co., mine run.	C. & E. I. No. 10030.	1.595	0.1355	7,380
July 6, 1905....	Brackett C. & C. Co., Keeler mine run.	C. & E. I. No. 12367	1.65	0.1236	8,091
July 28, 1905...	Kellyville Coal Co., washed pea.	C. & E. I. No. 6211.	1.50	0.1285	7,782
July 29, 1905...	....do...............	C. & E. I. No. 6211	1.50	0.119	8,403
Aug. 5, 1905 ...	Dering Coal Co., mine run.	C. & E. I. No. 25125	1.575	0.125	8,000
Aug. 7, 1905 ...	Dering Coal Co., Sullivan Co., screenings.	E. & T. H. No. 5132.	1.40	0.11	9,091
Aug. 8, 1905 ...	Consolidated Indiana Coal Co., Sullivan Co., screenings.	E. & T. H. No. 3239	1.35	0.105	9,524
Aug. 9, 1905 ...	Screenings...........	E. & T. H. No. 6534	1.30	0.0973	10,277
Aug. 11, 1905 ..	Ziegler, screenings...	I. C. No. 81184	1.50	0.1047	9,551

* See, "The Nature of True Boiler Efficiency," *Jour. Wes. Soc. Engrs.*, Oct., 1907, p. 677.

In nearly all stations the boilers must have sufficient overload capacity to take care of peak loads or to allow some of the boilers to be shut down for cleaning or repairs, since the installation of sufficient

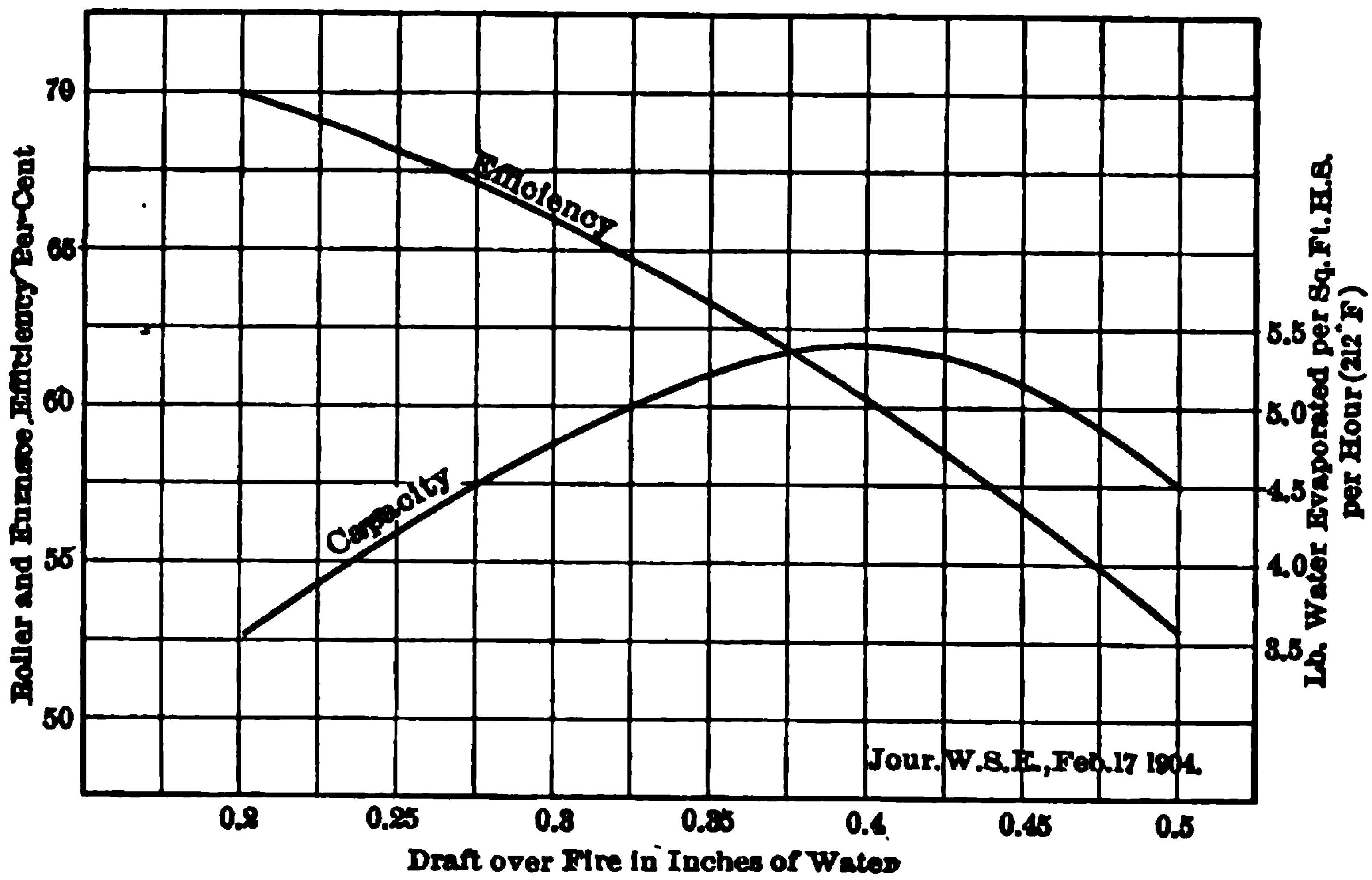


FIG. 40. Influence of Draft on the Efficiency and Capacity of a 350-Horse-power Babcock and Wilcox Boiler with Chain Grate.

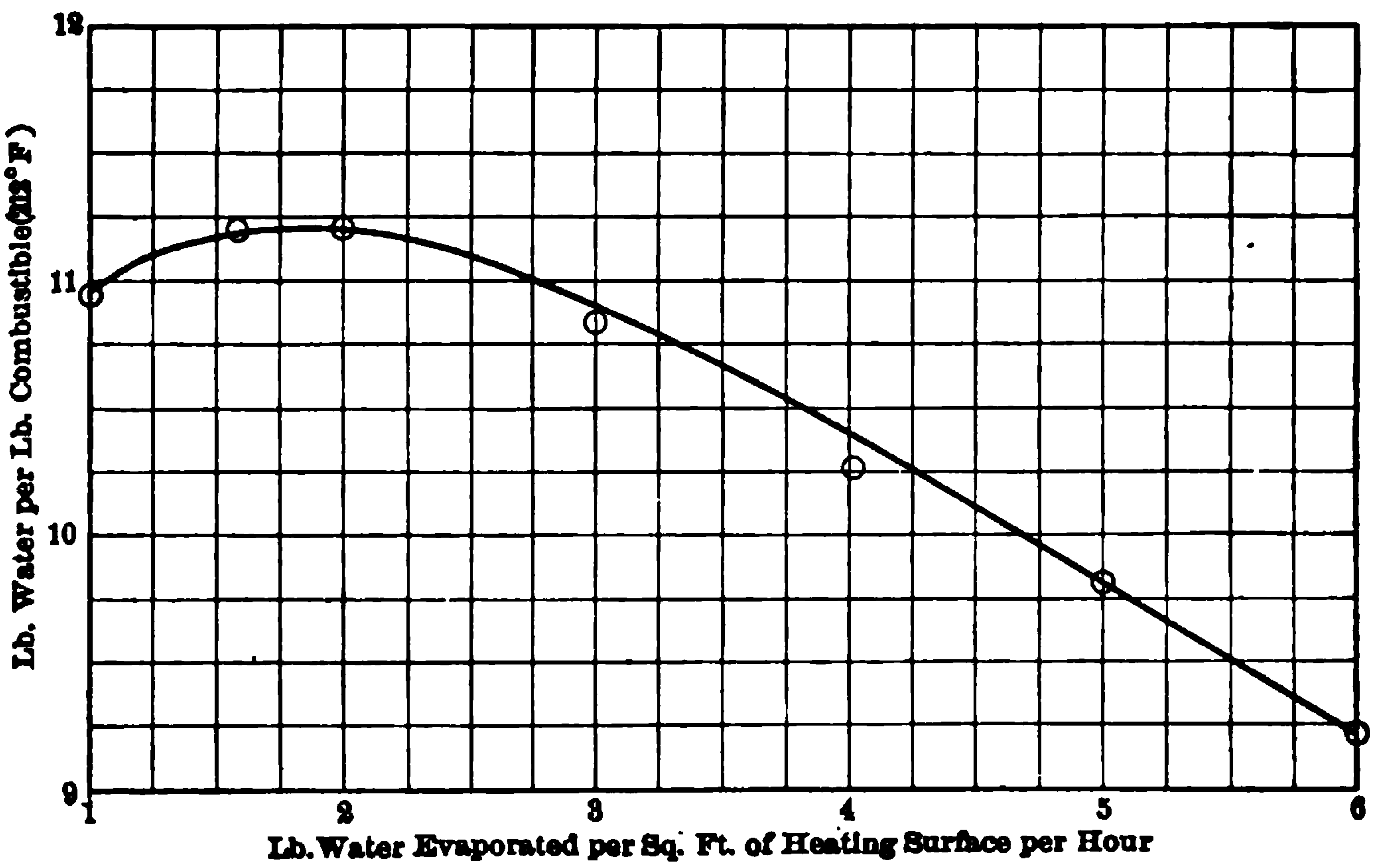


FIG. 41. Effect of Rate of driving on Economy of a 150-Horse-power Stirling Boiler, Hand Fired.

rated boiler capacity would be expensive and in many instances prohibitive in cost. In small stations, however, too large a boiler capacity frequently is to be preferred to an overloaded installation, since the

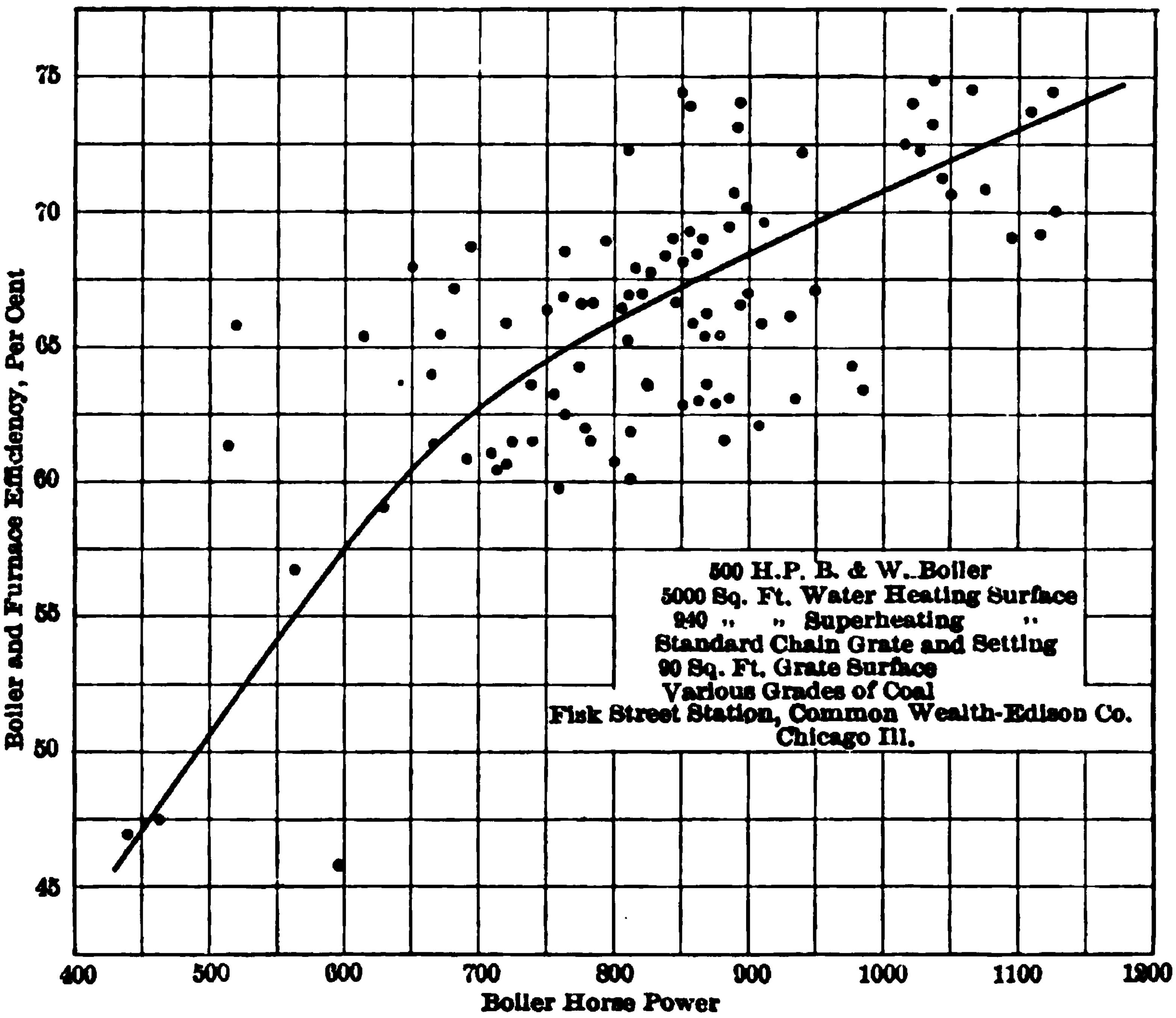


FIG. 41a. Relation Between Efficiency and Capacity, 500 H.P. Boiler, Fisk Street Station, Commonwealth Edison Co., Chicago.

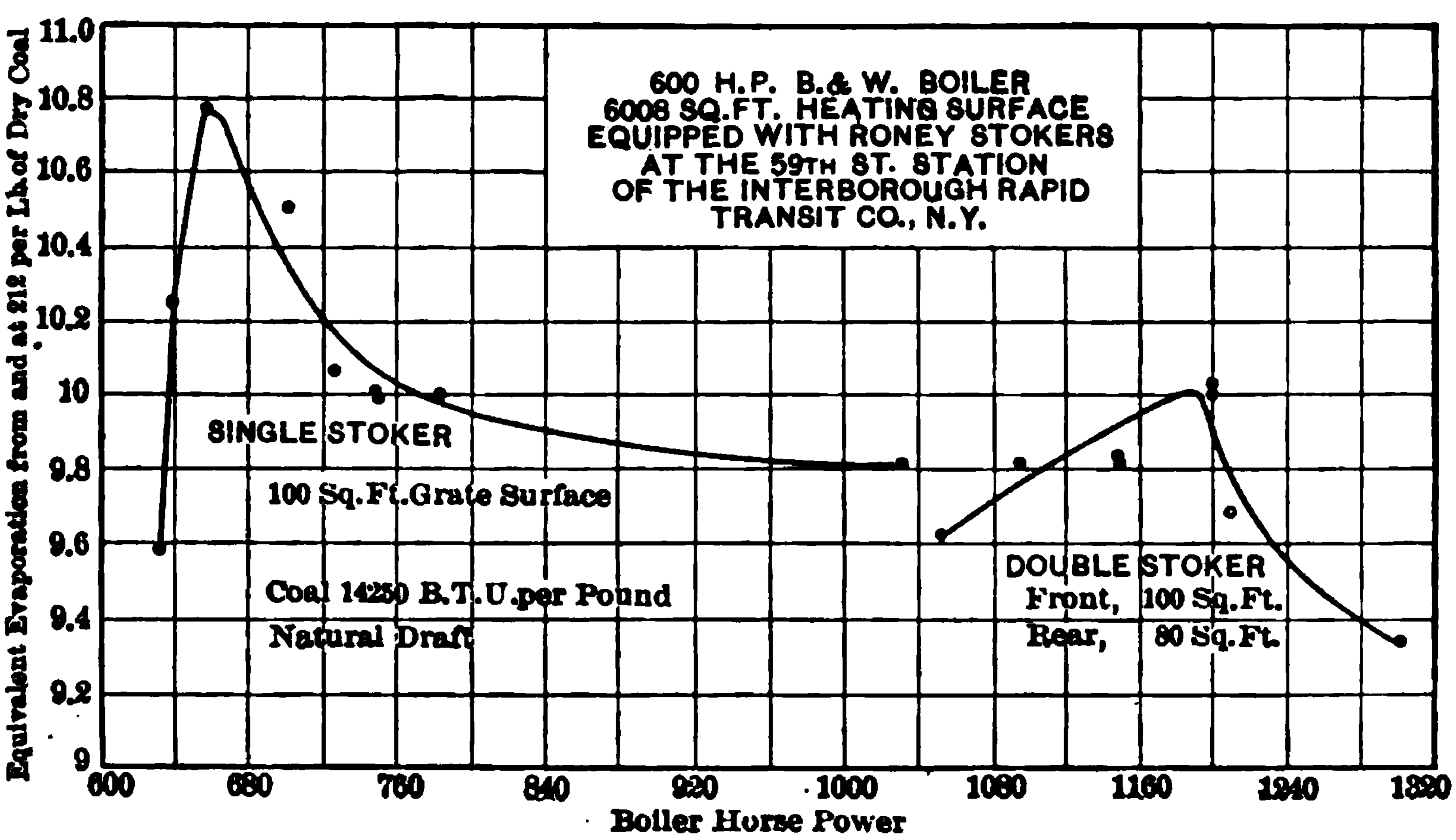


FIG. 41b. Effect of Rate of Driving on the Efficiency of a 600 H.P. B. & W. Boiler.

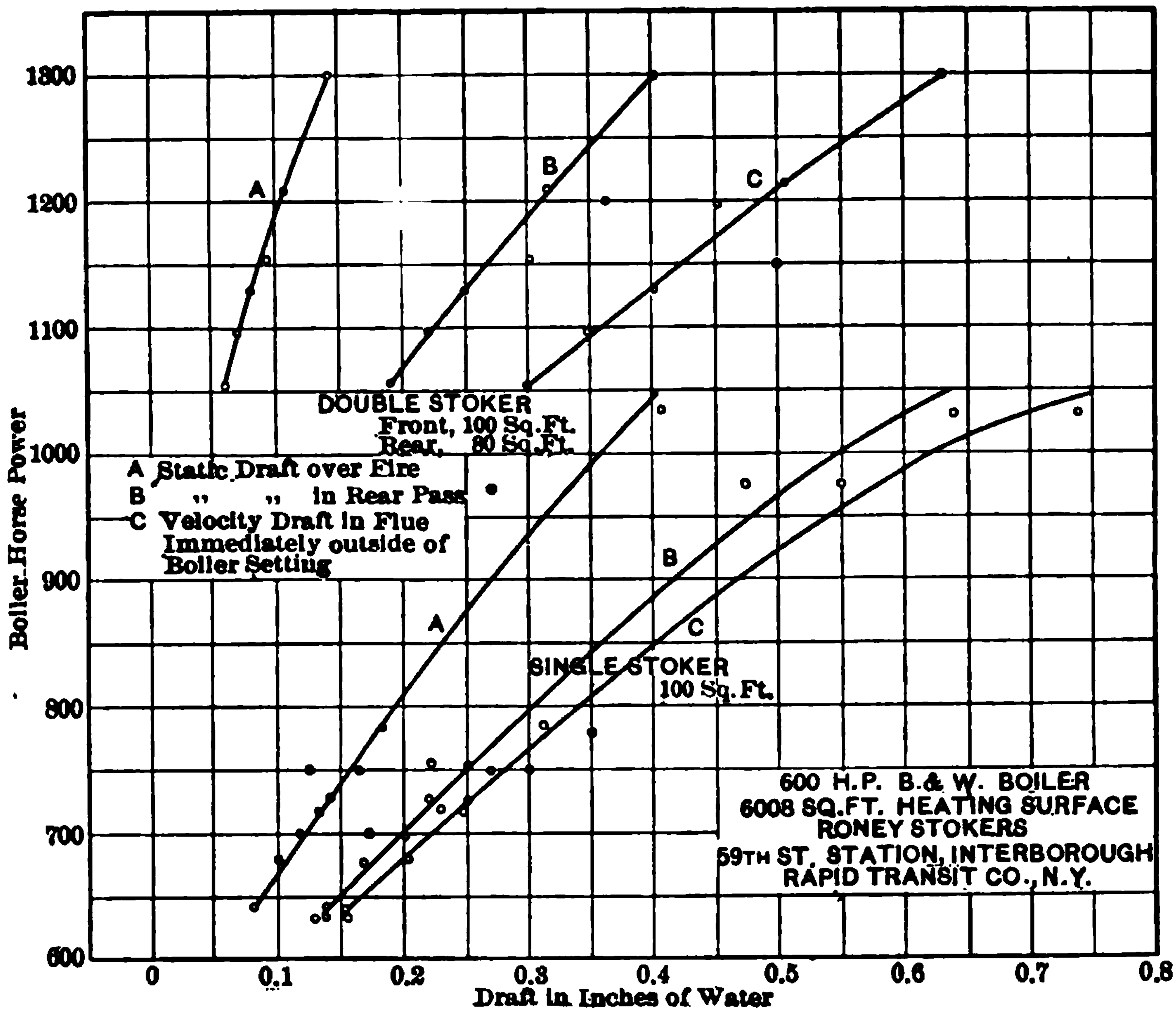


FIG. 41c. Influence of Draft on the Capacity of a 600 H.P. B. & W. Boiler.

TABLE 15a.

BOILER PERFORMANCE.

Pounds of Water Evaporated per Hour from and at 212 Deg. F. per pound of Fuel.

Calorific Value of Fuel, B.T.U. per Pound.	Boiler and Furnace Efficiency.									
	40	45	50	55	60	65	70	75	80	85
7,500	3.09	3.48	3.86	4.25	4.64	5.02	5.41	5.80	6.18	6.57
8,000	3.30	3.71	4.12	4.55	4.95	5.36	5.77	6.18	6.60	7.01
8,500	3.51	3.94	4.38	4.81	5.26	5.70	6.14	6.57	7.01	7.45
9,000	3.71	4.18	4.64	5.10	5.56	6.04	6.50	6.96	7.42	7.90
9,500	3.92	4.41	4.90	5.39	5.88	6.47	6.86	7.35	7.85	8.33
10,000	4.12	4.64	5.16	5.66	6.19	6.70	7.21	7.74	8.25	8.76
10,500	4.31	4.86	5.40	5.94	6.48	7.01	7.55	8.10	8.64	9.17
11,000	4.52	5.09	5.65	6.22	6.79	7.35	7.91	8.48	9.05	9.61
11,500	4.74	5.31	5.91	6.50	7.10	7.69	8.28	8.86	9.45	10.0
12,000	4.94	5.55	6.16	6.78	7.40	8.01	8.64	9.25	9.86	10.5
12,500	5.14	5.78	6.42	7.06	7.70	8.35	9.00	9.64	10.3	11.0
13,000	5.35	6.01	6.69	7.35	8.01	8.69	9.35	10.0	10.7	11.4
13,500	5.56	6.25	6.95	7.65	8.34	9.03	9.72	10.4	11.1	11.8
14,000	5.75	6.48	7.20	7.91	8.64	9.35	10.1	10.8	11.6	12.2
14,500	5.96	6.70	7.45	8.20	8.95	9.70	10.5	11.2	12.0	12.7
15,000	6.18	6.95	7.72	8.50	9.26	10.1	11.8	11.6	12.4	13.1

extra first cost of the former may be less than the loss due to poor efficiency and depreciation in the latter.

As far as forcing is concerned the fire-tube boiler is as effective as the water-tube, more depending upon the furnace, grate surface, draft and character of fuel than upon the type of boiler. All boilers are subject to more or less priming at heavy overloads, and the overload capacity is often limited on this account.

The Forcing Capacity of Fire-Tube Boilers: F. W. Dean, Trans. A.S.M.E., 26–92.
Increasing Capacity of Steam Boilers: Kreisinger and Ray, Power, May 24, 1910.

76. Thickness of Fire. — For a given furnace and boiler, quality and size of fuel and intensity of draft, a certain depth of fuel will give maximum efficiency. Too thin a fire results in an excess of air and too thick a fire in a deficiency, the economy being lowered in either case. On account of the number of conditions upon which the proper thickness depends, it can only be determined for a particular case by actual test, the available data being insufficient for drawing conclusions. The curves in Fig. 42 are plotted from a series of tests made on a 350-horse-

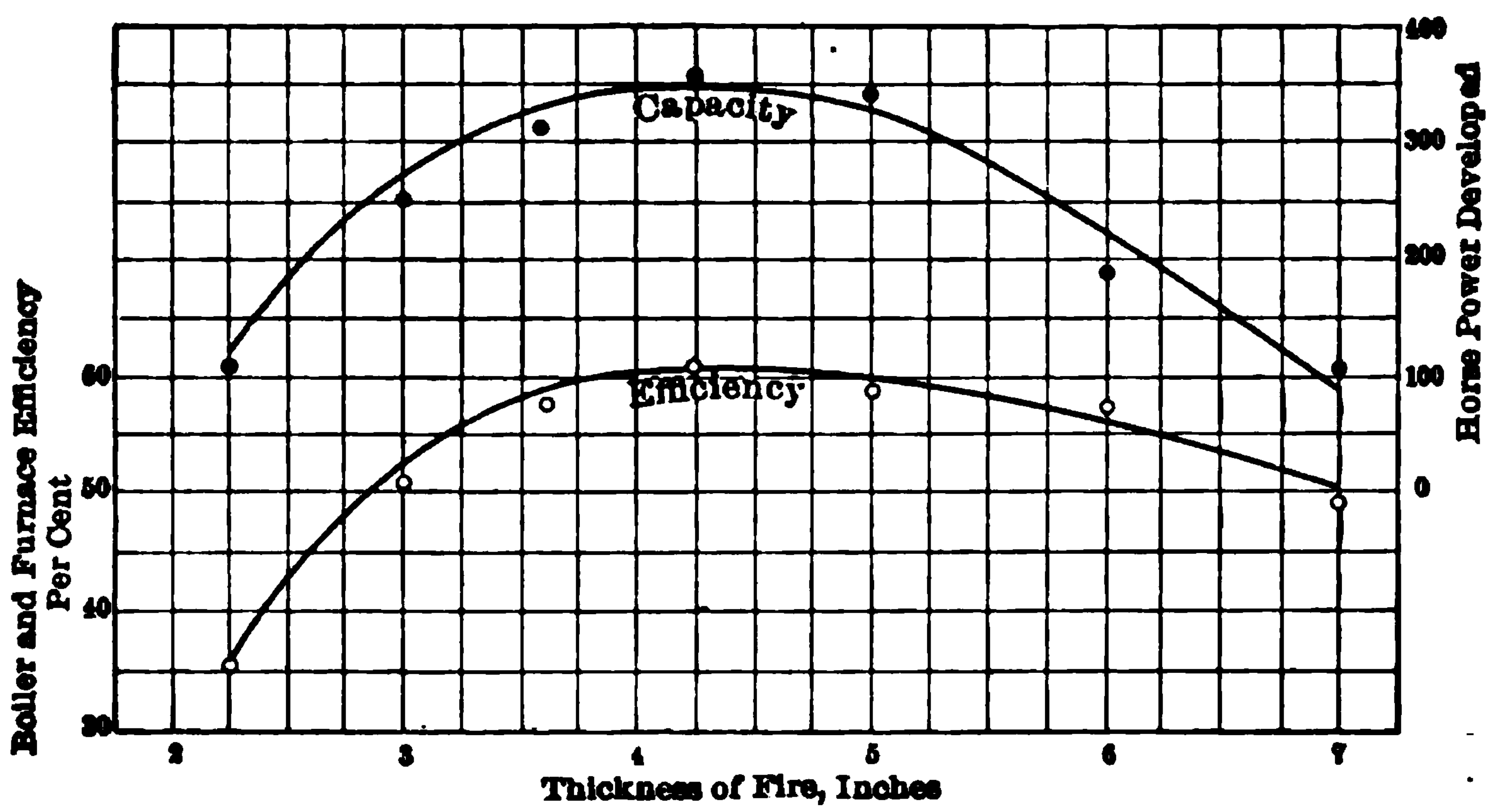


FIG. 42. Effect of thickness of Fire on the Capacity and Efficiency of a 350-Horse-power Stirling Boiler, equipped with Chain Grate.

power Stirling boiler equipped with chain grate at the power plant of the Armour Institute of Technology. The damper was left wide open throughout the test and the speed of the grate kept constant. Ratio of grate to heating surface, 1 to 42. Carterville washed coal No. 4 was used in all tests. The curves in Fig. 43 refer to the performance of a 150-horse-power water-tube boiler equipped with chain grate at the University of Illinois Engineering Experiment Station at Urbana.

The curves in Fig. 44 are plotted from a series of tests on a 500-horsepower Babcock & Wilcox boiler equipped with chain grate at the Fisk Street station of the Commonwealth Edison Company, Chicago, Ill. In these tests the conditions of operation are not exactly comparable, but they serve to show the variation of economy with thickness of fire in each case. In general, with natural draft, fine sizes of coal necessitate thin fires, since they pack so closely as to greatly restrict the draft. Thin fires require closer attention to prevent holes being burned in

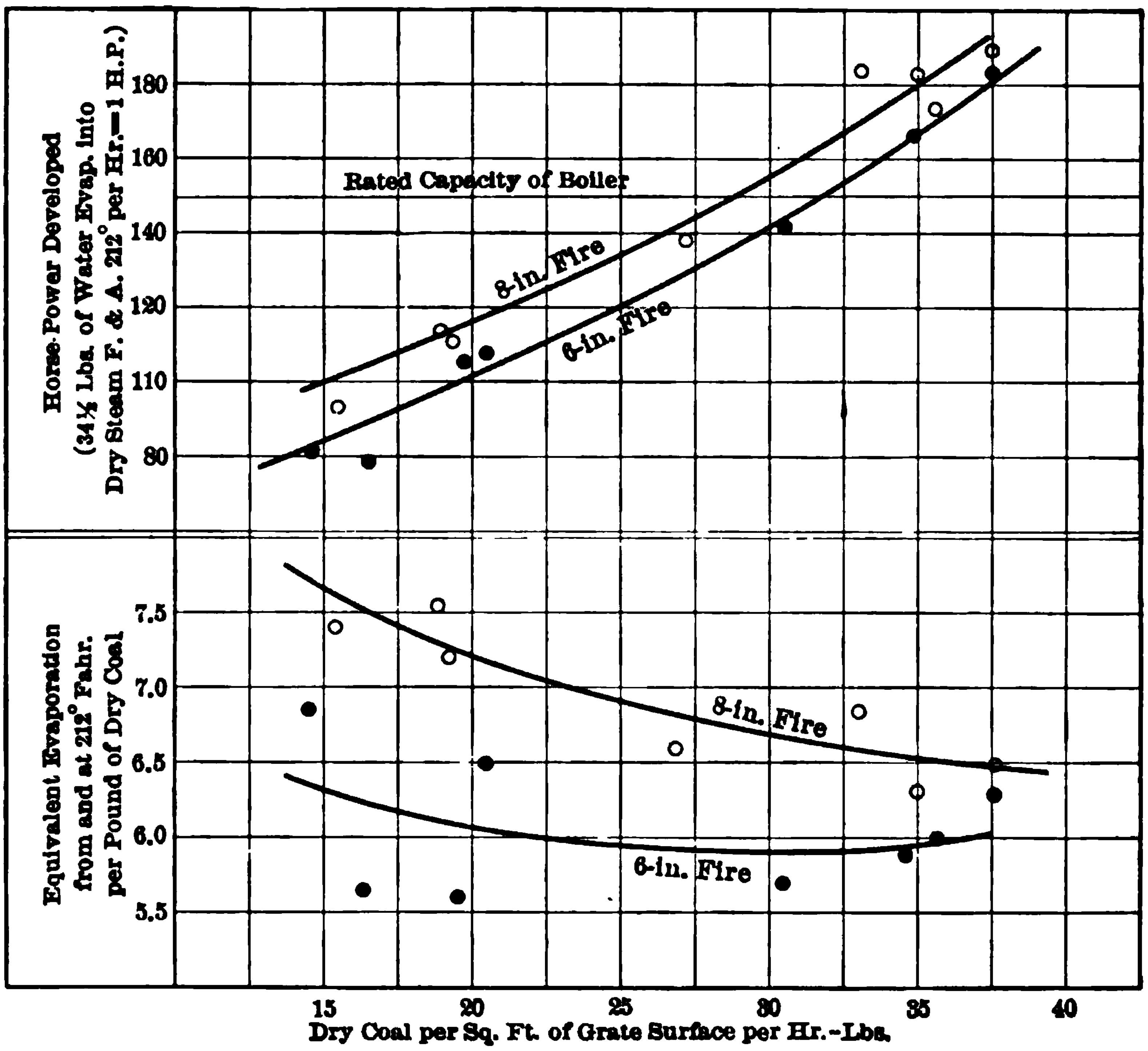


FIG. 43. Effect of Thickness of Fire on the Capacity and Efficiency of a 150-Horse-power Water-Tube Boiler.

spots, and respond less readily to sudden demands for steam, but have the advantage of letting the air required pass through the grate, whereas thick fires often require air to be supplied above the grate to insure complete combustion. Thick fires require less attention and hence are preferred by firemen. Where sufficient draft is available thick fires are more efficient than thin ones, as the air excess is more readily controlled.

77. Influence of Initial Temperature on Efficiency. — In general the higher the initial temperature of the furnace the greater will be the efficiency of the *heating surface*, since the heat transmitted varies almost directly with the difference of temperature between the water and the products of combustion. If the heating surface is properly distributed so that the final temperature of the escaping gas remains constant, the efficiency of the boiler and furnace will increase as the initial temperature increases, though not in direct proportion. This is on the assumption

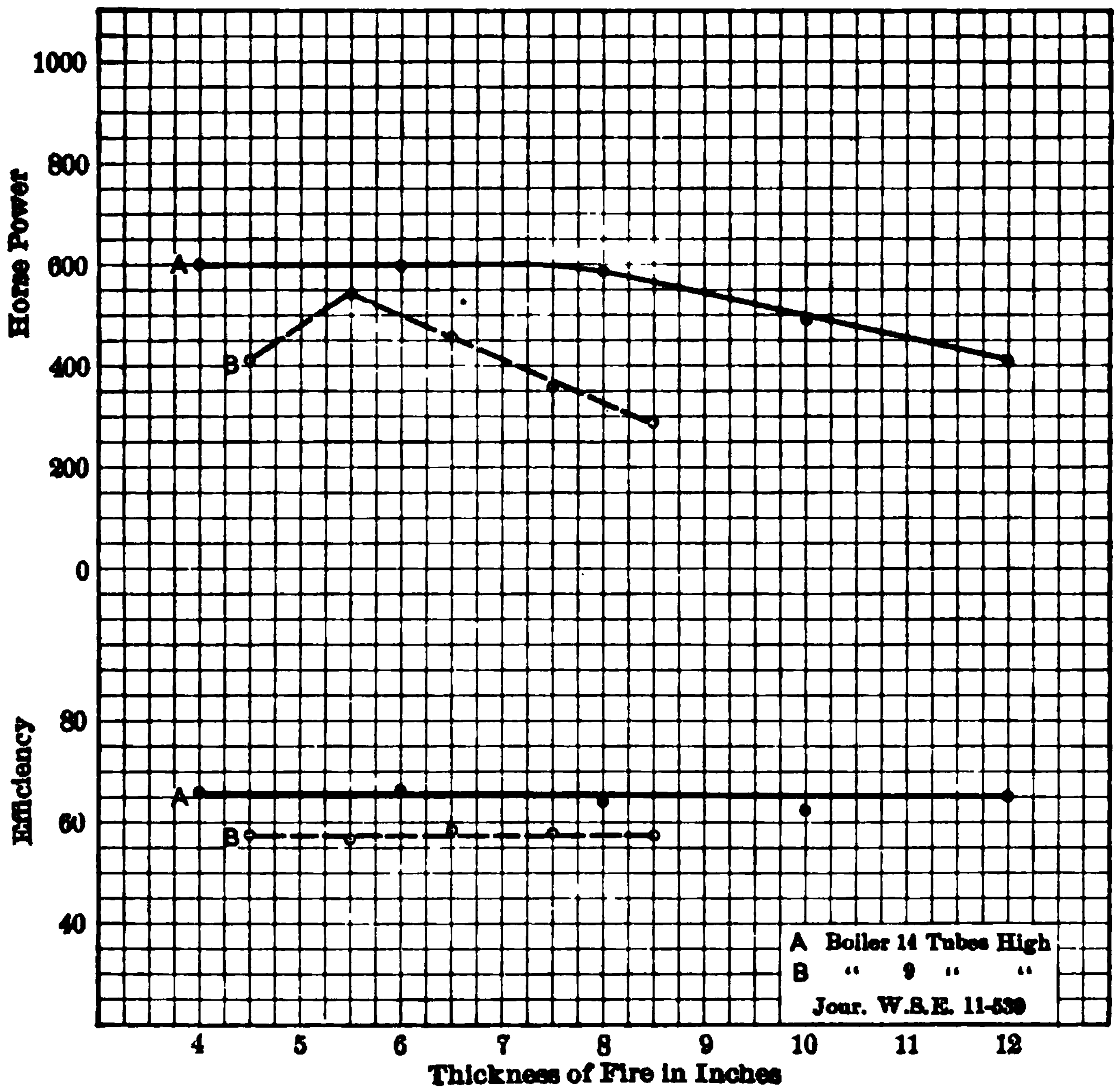


FIG. 44. Effect of Thickness of Fire on the Capacity and Efficiency of a 500-Horse-power Babcock and Wilcox Boiler.

that the amount of heat generated per hour is the same throughout all ranges in temperatures. With a condition where the amount of heat generated remains constant and the initial temperature varies, the final temperature of the escaping gases remains practically constant, and in such cases high initial temperatures are productive of high *boiler* and *furnace efficiencies*. In practice these conditions are seldom realized and high furnace temperatures are not necessarily productive of high boiler and furnace efficiencies. Some tests show a decided gain in efficiency with the higher furnace temperatures ("Some Perform-

ances of Boilers and Chain Grate Stokers, with Suggestions for Improvements," A. Bement, *Jour. West. Soc. Engrs.*, February, 1904), and others show little if any improvement (" A Review of the United States Geological Survey Fuel Tests under Steam Boilers," L. P. Breckenridge, *Jour. West. Soc. Engrs.*, June, 1907). The majority of high efficiency records, however, are associated with high furnace temperatures.

78. Cost of Boilers and Settings. — Figures giving the cost of boilers, irrespective of type, at so much per horse power are misleading, since the cost does not increase in the same ratio as the power. The wide variation in cost on the horse-power basis is partly due to the difference in rating. For instance, Scotch marine boilers are ordinarily rated at 8 square feet of heating surface per horse power and return tubular boilers at 12 square feet. The price approximates one dollar per square foot of heating surface for all boilers over 100 horse power. The cost of water-tube and fire-tube boilers may be roughly estimated by the following formulas (C. H. Benjamin, Engr. U.S., Nov. 15, 1902):

(A) Cost in dollars = 500 + 9.2 × rated horse power. (16)

(B) Cost in dollars = 500 + 8.5 × rated horse power. (17)

(C) Cost in dollars = 100 + 6.5 × rated horse power. (18)

(D) Cost in dollars = 100 + 5.0 × rated horse power. (19)

(A) Horizontal water-tube boilers, 125 pounds pressure, 10 square feet heating surface per horse power.

(B) Vertical water-tube boilers, other conditions as in (A).

(C) Horizontal return tubular boilers, 12 square feet heating surface per horse power.

(D) Small vertical fire-tube boilers.

The cost of Scotch-marine boilers rated on a basis of 8 square feet per horse power may be estimated by means of formula (A).

The cost of plain settings may be roughly approximated as follows:

Horizontal water tube:

$$\text{Cost} = 400 + 0.8 \times \text{rated horse power.} \tag{20}$$

Return tubular:

$$\text{Cost} = 300 + 0.7 \times \text{rated horse power.} \tag{21}$$

79. Selection of Type. — Boilers constructed by builders of good repute are usually designed for safety, durability, and capacity, and rigid specifications and inspection of material and workmanship are ordinarily not necessary, as the makers' reputations are sufficient guarantee of their worth. Marked departure from standard designs must necessarily be specified, but in most cases instructions are limited to the extent of heating and grate surface, the character of the furnace,

and arrangement of setting. Numerous tests on various types of boilers show practically the same efficiency provided the furnaces and boilers are properly designed, so that the relative merits may be considered with reference to (1) durability; (2) accessibility for repairs; (3) facility for cleaning and inspection; (4) space requirements; (5) adaptability to the type of furnace and stoker desired; (6) capacity; and (7) cost of boiler and setting. For high pressure, 150 pounds per square inch or more, the water-tube or some form of internally fired boiler in which the shell plates are not exposed to the high temperature of the furnace is considered safer than the horizontal tubular boiler because the shell plates and the seams of the latter must be of considerable thickness in large units, and being exposed to the hottest part of the fire are likely to give trouble, especially if the water contains scale or sediment-forming elements. Return tubular and stationary locomotive boilers are seldom made in sizes over 200 horse power and hence are not to be considered for large units. For sizes over 150 horse power where overhead room is limited the return tubular boiler is most commonly installed, unless high pressure is essential, in which case the internally fired Scotch-marine boiler is peculiarly adaptable. The water-tube boiler is usually employed in large central stations for high-pressure units of 300 to 500 horse power. The particular type of water-tube boiler is to some extent a matter of personal taste on the part of the engineer. For small powers and for intermittent operation, small vertical or horizontal fire-box boilers have the advantage of low first cost. The small air leakage and radiation losses give internally fired boilers an advantage over externally fired fire-tube or water-tube types, but this is partly offset by the greater extent of regenerative surface in the setting of the latter.* Internally fired boilers are more expensive than the externally fired, though the extra cost of setting and foundation in the latter may bring the total cost of the entire equipment to practically the same figure. The design and installation of the boilers and furnaces should be left at the outset to a capable engineer.

Makers usually request the following information from intending purchasers:

1. Steam pressure desired.
2. The quantity of steam demanded.
3. The kind of fuel to be burned.
4. The type of furnace or stoker.
5. The nature and intensity of draft.
6. Nature of setting.
7. Probable temperature of the feed water.

* At the power plant of the Cosmopolitan Electric Co., Chicago, the brick settings of the boilers (500 h.p. B. & W.) are completely encased with riveted boiler plates.

The complete specifications for a return tubular boiler are given in Chapter XIX.

Choice of Boilers for Steam Power Plants: Am. Elecn., June, 1900, p. 261, Dec., 1905, p. 633, May, 1901, p. 217, June, 1903, p. 256; Engr., Lond., Sept., 1898, p. 232, May 29, 1903, p. 555; St. Ry. Rev., Oct., 1901, p. 789; Elec. Rev., Lond., June 16, 1899, p. 973; St. Ry. Jour., Oct. 5, 1901, p. 453, Oct., 1905, p. 731; Mech. Eng., May 16, 1903; Min. Rept., Feb. 21, 1907.

80. Grates. — Grates may be divided into three general classes, namely, stationary, rocking, and traveling grates. The latter are treated in Chapter IV. Stationary grates are generally made of cast-iron sections in a variety of shapes as illustrated in Fig. 45. The bars are ordinarily from 3 inches to 4 inches deep at the center (this makes them strong enough to carry the load caused by the weight of the fuel without sagging even when the top is red hot), $\frac{3}{4}$ inch wide at the top, and taper to $\frac{3}{8}$ inch at the bottom to enable the ashes to drop clear. The width of the air space is determined by the size of the fuel to be used, the average proportions being given in Table 16.

TABLE 16.

AIR SPACES AND THICKNESS OF GRATE BARS.

Size and Kind of Coal.	Width of Air Spaces.	Thickness of Grate Bars.
	(Inch)	(Inch)
Screenings	$\frac{1}{4}$	$\frac{3}{8}$
Anthracite —		
Average	$\frac{1}{2}$	$\frac{3}{4}$
Buckwheat	$\frac{3}{8}$	$\frac{3}{8}$
Pea or nut	$\frac{1}{2}$	$\frac{1}{2}$
Stove	$\frac{5}{8}$	$\frac{1}{2}$
Egg	$\frac{3}{4}$	$\frac{1}{2}$
Broken	$\frac{7}{8}$	$\frac{1}{2}$
Lump	1	$\frac{3}{4}$
Bituminous, average	$\frac{5}{8}$	$\frac{3}{4}$
Wood —		
Slabs	$\frac{3}{4}$	$\frac{3}{4}$
Sawdust	$\frac{1}{4}$ to $\frac{3}{8}$	$\frac{3}{8}$
Shavings	$\frac{1}{2}$ to $\frac{5}{8}$	$\frac{1}{2}$

The "Tupper" and "Herringbone" grate bars are stiffer and less likely to warp than the common form, but are not so readily sliced and therefore not so convenient with coal that clinkers badly. Sawdust or pin-hole grates are used in burning sawdust, tan bark, and very small sizes of coal. Grates are often set horizontally and the bars are held in place simply by their own weight, but long grates are best placed sloping toward the rear to facilitate firing. The front of the grate when

designed for bituminous coal is often made solid, this portion being called the " dead plate." It serves to hold the green fuel until the hydrocarbons have been distilled off, when the charge is pushed back on the open grate at the time of next firing. The length of a single bar or casting should not exceed three feet. The length of grate may be

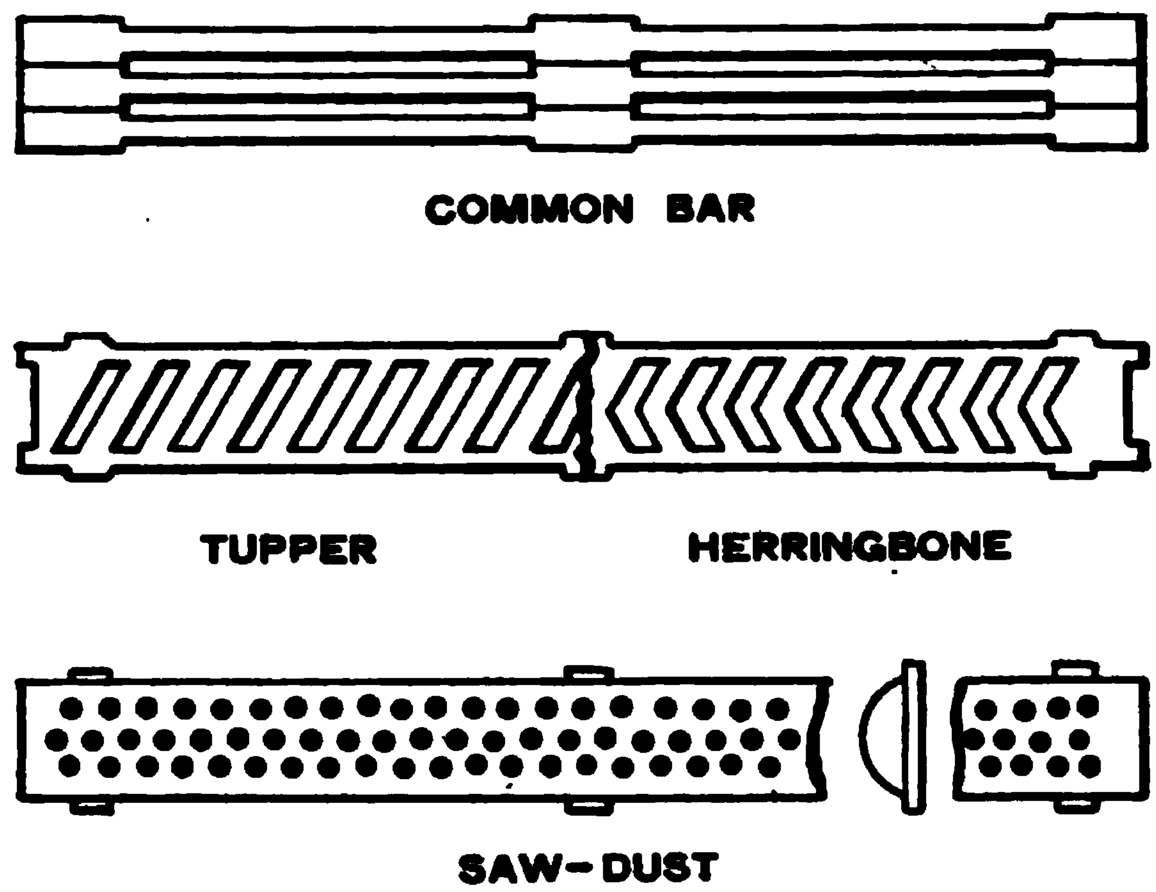


FIG. 45. Types of Grate Bars.

made of two or three bars and should not exceed 6 feet with bituminous coal, as this is the greatest length of fire that can be readily worked by a stoker. With buckwheat anthracite furnaces 12 feet in depth are not unusual, as anthracite fires require no slicing.

The disadvantage of using stationary grates is that the fire is not easily cleaned. Unless the air spaces are kept free of clinkers and ashes, combustion is hindered and the fire rendered sluggish. Frequent cleaning, however, is wasteful of fuel and reduces the furnace efficiency by letting in a large excess of air every time the fire door is opened.

81. Rocking Grates. — Shaking grates have the advantage of permitting stoking without opening the fire door and require less manual labor than stationary grates. There is a great variety of sectional shaking grates on the market and some of them are made self-dumping. One of the best known types is illustrated in Fig. 46. Each row or section of grate bars is divided into a front and a rear series by twin stub levers and connecting rods. An operating handle is adapted to manipulate either one or both of the levers in such a manner that the front and rear series may operate separately or together. The shaking movement causes no increase in the size of the openings and hence prevents the waste of fine fuel. Ordinarily the width of the grate is made

equal to two or more rows of grate bars so that the live fire may be shoved sidewise from one row to the other when cleaning. A depth of

FIG. 46. A Typical Rocking Grate.

fire of from 6 to 10 inches is carried according to the nature of the fuel and the available draft.

Grate Bars: Engr. U.S., Nov. 1, 1906, p. 728, Jan. 1, 1907, p. 68; Am. Elecn., Jan., 1904, p. 269.

82. Blow-Offs. — Boilers must be provided with blow-off pipes for draining off the water and for discharging sediment- and scale-forming material. The "bottom blow" is ordinarily an extra heavy pipe of suitable diameter connected to the lowest part of the boiler and fitted

FIG. 47. Horizontal Blow-off Connection to Head.

FIG. 48. Vertical Blow-off Connection to Shell.

with a valve or cock, or both. (See paragraph 349.) Fig. 47 shows an arrangement of horizontal blow-off connected to the head of a return tubular boiler and Fig. 48 a vertical blow-off connected to the shell.

The latter is the better arrangement. The blow-off pipe where it passes through the back connection is covered with magnesia, asbestos, or fire brick. When exposed to the action of extremely hot furnace gases as in forced draft installations, the arrangement illustrated in Fig. 49 is sometimes used to prevent the pipe from burning out. When the blow-off cock is shut and the valve on the vertical branch open there is a continuous circulation of water. Where boilers are arranged in batteries, the battery may have a common outlet for the blow-off

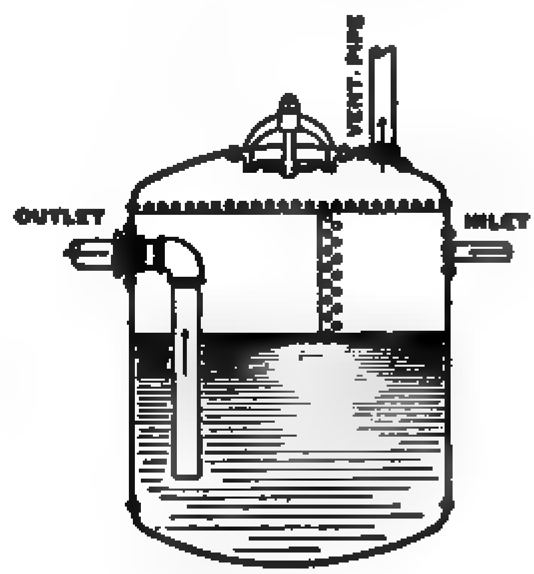


FIG. 49. Blow-off Connection with Circulating Pipes.

FIG. 50. Blow-off Tank and Connections.

pipes, as illustrated in Fig. 388. Usually the blow-off pipes may discharge into the open air, but this is not permissible in large cities, nor is it lawful to blow directly into the sewer. In this case the water and sediment may be discharged into a *blow-off tank* and permitted to cool before delivery to the sewer, as illustrated in Fig. 50.

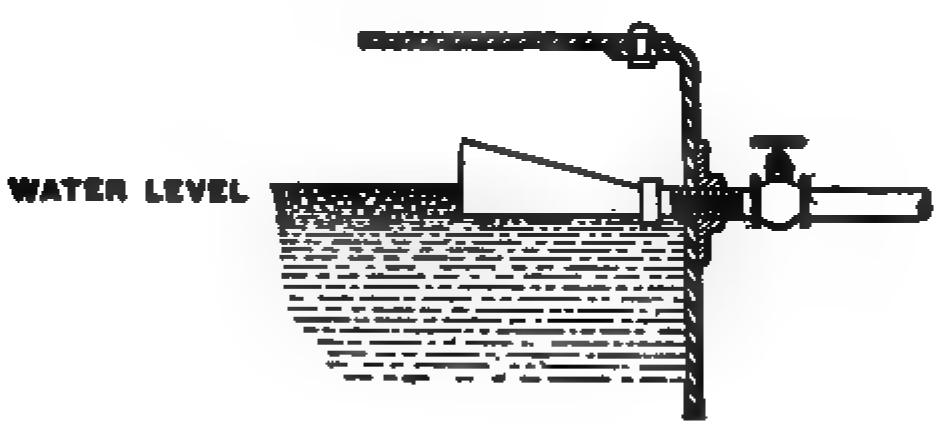


FIG. 51. Surface Blow-off.

"*Surface blows*" are often installed to remove scum, grease, and floating or suspended particles of dirt. The bell-mouthed shape shown in Fig. 51 permits the skimmer to accommodate itself to varying water

level, and it is sometimes provided with a float and with a flexible joint, Fig. 52.

83. Damper Regulators. — For maximum furnace efficiency the draft must be regulated to burn just enough fuel to supply the steam

Fig. 52. Buckeye Skimmer.

required. Where forced draft is employed this is done by regulating the speed of the blower. With natural draft it is the usual practice to regulate the draft by means of dampers placed in the uptake, and in order that the regulation may be effective it should be automatic. Automatic dampers are economical and useful and are particularly desirable in plants where the demand for steam fluctuates rapidly. There are several successful types on the market, some operated by water pressure and others by direct boiler pressure. Fig. 53 illus-

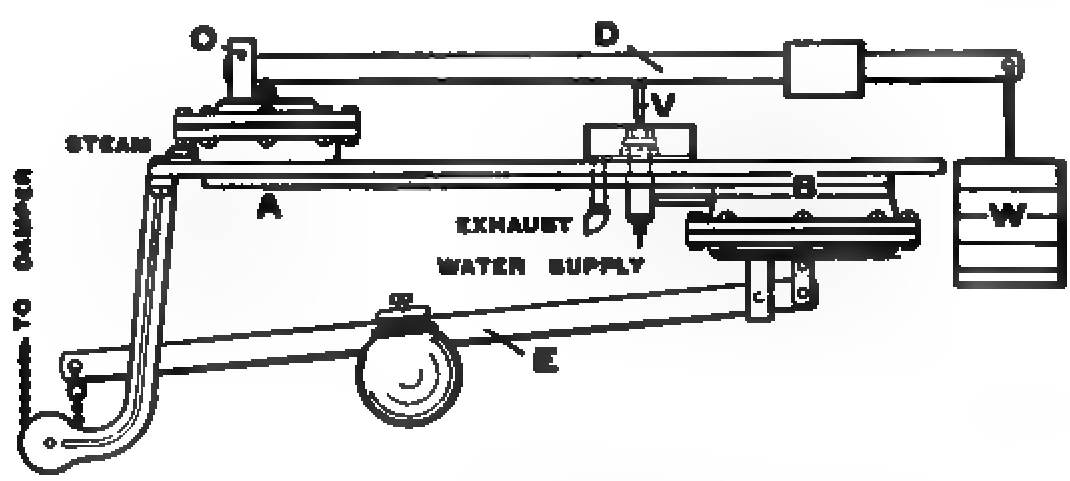


Fig. 53. Kitts Hydraulic Damper Regulator.

trates such a mechanism. Full boiler pressure acting at all times on a diaphragm A raises or lowers a weight W attached to arm D according as the steam pressure increases or decreases. Arm D actuates a small valve V which controls the supply of water to chamber B. A diaphragm in chamber B raises and lowers the damper as the water pressure varies, a drop of 0.5 pound being sufficient to open the damper to its maximum.

The steam diaphragm has a movement of only 0.01 inch and the water diaphragm 0.5 inch. When properly adjusted and given proper attention automatic dampers work in a very satisfactory manner.

Fig. 54 shows a section through the Tilden damper regulator, illustrating the principles of the steam-actuated type. The device is con-

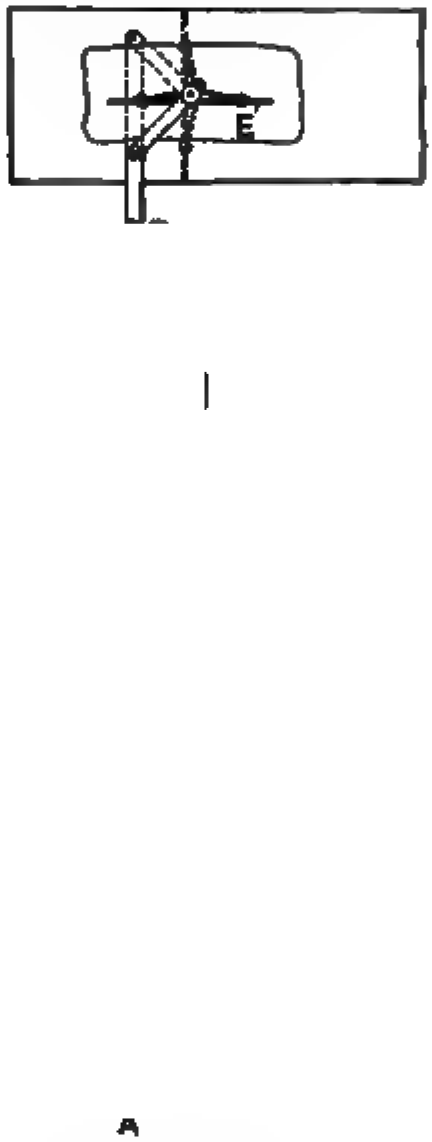


Fig. 54. Tilden Steam Actuated Damper Regulator.

Fig. 55. Simple Water Column.

nected directly to the boiler by pipe A. The pressure on piston B is balanced by spring C under normal conditions of operation. Any variation from the normal will cause the rod R to move up or down, so that the dampers are opened or closed in proportion to the change in pressure. The chamber N is separate from chamber M, so that steam or water cannot come in contact with the spring. Piston D acts as a guide only.

Damper Regulators: Engr. U.S., Jan. 1, 1907, p. 58; Elec. Wld., May 2, 1908.

84. Water Gauge. — The water level in a boiler is usually indicated either by a gauge glass, by try cocks, or both, connected directly to the boiler as in Fig. 1, or to a *water column* or *combination* as in Fig. 55.

Each gauge-glass connection should be fitted with a stop valve which may be closed in case the tube breaks. In large boilers these valves, usually of the quick-closing type, are conveniently operated from the boiler-room level by means of a chain attached to the valve stem. Self-closing automatic valves are frequently employed, one type being illustrated in Fig. 56. If the glass breaks the outrush of steam forces the ball against a conical seat and shuts off the supply. When a new glass is inserted the ball is forced back by slowly screwing in the valve stem. Hinged valves mechanically operated from without are considered more reliable than ball valves, as scale is less likely to render them inoperative.

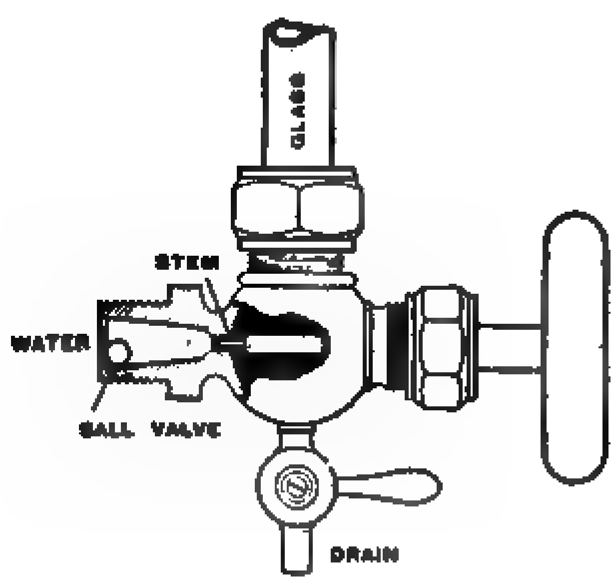


FIG. 56. Water Guage with Self-closing Valve.

WATER

FIG. 57. Combined Water Column and High and Low-Water Alarm.

Try cocks or *gauge cocks* are set at points above and below the desired water level, preferably connected directly to the boiler shell but sometimes to a water column as in Fig. 55. The water level is ascertained by opening the cocks in succession.

The objection to the latter arrangement is that accident to or a stoppage of the piping renders both gauge glass and try cocks useless. Water columns should be blown out once a day and the gauge cocks opened to see that the height of the water indicated tallies with that shown by the glass. Some engineers prefer two separate columns to each boiler and no cocks, others rely solely upon cocks.

The water column shown in Fig. 57 has an alarm whistle, controlled by two floats, which gives a high and low-water alarm. Numerous devices of this class are on the market but they are usually regarded as

unreliable and most engineers are content to depend upon water gauge and try cocks.

Water Gauges and Columns: Mach., Sept., 1905, p. 31; Power, Aug., 1905, p. 483; Am. Elecn., July, 1904, p. 359; Engr. U.S., Jan. 1, 1907, p. 58.

85. Fusible or Safety Plugs. — Fusible or safety plugs as illustrated in Fig. 58 are brass plugs provided with a fusible metal core. They

Fig. 58. Types of Fusible Plugs.

are inserted in the shell or tubes at the lowest permissible water line. When covered by water the heat is conducted away sufficiently fast to keep the temperature below the fusing point, but when uncovered the low conductivity of the steam prevents the rapid withdrawal of heat, whereupon the alloy melts and the blast of escaping steam gives warning. The melting point of fusible metals being sometimes uncertain, plugs occasionally blow out without apparent cause and at other times fail to act when shell is overheated. Fusible plugs are required by law in many cities.

86. Mechanical Tube Cleaners. — Although purifying plants, boiler compounds, and the like are preventive of scale formation to a great extent, experience shows that the most satisfactory method is to use mechanical tube cleaners for cutting or breaking the scale. The principles of construction of these devices vary widely according to the types of boilers in which they are used, and depend upon the nature of the

Fig. 59. Mechanical Tube Cleaner, — Hammer Type.

duty which they must perform. They may be conveniently divided into two classes:

1. Those which loosen the scale by a series of rapid hammer blows, Fig. 59.
2. Those which cut out the scale by a revolving tool, Fig. 60.

The hammer device is applicable to either the water- or fire-tube type of boiler, but the revolving cutter is applicable to the water-tube only. Steam, compressed air, or water under pressure may be used as the motive power, though the latter is the most convenient and satisfactory.

Referring to Fig. 59, the hammer head J is given a rapid motion, which may reach 1,500 vibrations per minute, and subjects the tube to repeated shocks, thereby cracking the brittle scale and jarring it loose from the water surface of the tube. The cleaner head is attached to a flexible pipe of sufficient length to enable it to be pushed from one end to the other. Even if carefully manipulated the hammer is apt to injure the tube by swaging it to a larger diameter, producing crystallization in the metal and causing leaks where the tubes are expanded into the sheets, hence its use is not to be recommended.

Hydraulic turbine cutters are made in many designs, one of which is shown in Fig. 60. The cylindrical casing D contains a hydraulic

Fig. 60. Mechanical Tube Cleaner, — Turbine Type.

turbine consisting of a fixed guide plate which directs the water at the proper angles upon the vanes of the turbine wheel T. The cutters C revolve at high speed and chip the scale into small pieces. The stream of water flowing from the turbine envelops the cutters, keeps their edges cool, and washes away the scale as fast as it is detached. Different styles of cutter wheels are furnished with each cleaner so as to adapt the device to all kinds of scale formations. In well-managed plants scale is not permitted to deposit to a thickness greater than $\frac{1}{16}$ to $\frac{3}{16}$ of an inch.

The soot and cinders which accumulate on the inside surface of fire-tube boilers are removed by mechanical scrapers, brushes, or steam-jet blowers. (For a description of these devices see *American Electrician*, April, 1904, p. 576.)

The tubes of a water-tube boiler are cleaned externally by means of a steam jet.

Boiler Accessories: Am. Elecn., April, 1903, p. 194, Feb., 1905, p. 67, June, 1904, p. 269, July, 1904, p. 339; Am. Mach., April 21, 1901, p. 518; Engr. U.S., Jan., 1907, p. 56.

Boiler Arches: Boiler Maker, Aug., 1907.

Blow-off Connections: Locomotive, Oct., 1906; Elec. World, Nov. 2, 1907; Nat. Engr., June, 1904; Eng. Rec., May 9, 1908.

Bracing: Boiler Maker, April, 1905; Mach., Sept., 1903, p. 18, Oct., 1903, p. 83; Power, Jan., 1903, p. 24, Oct., 1905, p. 611, Nov., p. 687; Eng. News, Dec. 15, 1904, p. 533; Trans. A.S.M.E., 18–989; Am. Mach., June 3, 1897, June 2, 1898, p. 404; Engr. U.S., Jan. 1, 1907, p. 18; Engr., Lond., April 25, 1900, pp. 412, 419; Prac. Engr., Jan., 1907.

Boiler Cleaning: Am. Elecn., Dec., 1900, April, 1904, p. 174; Power, May and Oct., 1905, Aug., 1906, p. 465; Locomotive, Oct., 1904; Boiler Maker, Aug., 1905; Engr. U.S., Jan. 1, 1907, p. 109.

Boiler Design: Engr. U.S., Jan. 15, 1902, p. 59; Eng. Mag., May, 1904, p. 233; Eng. Rec., July 14, 1900, May 18, 1901, p. 467, Oct. 12, 1901, p. 347; Power, Oct., 1901, p. 14, March, 1906, p. 147; St. Ry. Rev., Feb. 15, 1899, p. 125; West. Elecn., April 20, 1901, p. 267; Am. Mach., April 21, 1904; Mach., Oct., 1902.

Boiler Dimensions: All Types of Stationary Boilers: Eng. U.S., Jan. 1, 1907, p. 10, Aug. 1, 15, 22, 1903. *Small Marine Boilers:* Am. Mach., Sept. 3, 1896, p. 823. *Tubular Boilers:* Mach., Oct., 1902, p. 94.

Circulation in Boilers: Eng. Rec., July 20, 1901; Cassier's Mag., Jan., 1905; Elec. Rev., Lond., April 4, 1902; Engng., April 18, 1902; Engr., Lond., Nov. 6, 1903; Am. Mach., Jan. 14, 1897, p. 40, Sept. 20, 1900, p. 910; Eng. News, Jan. 18, 1900, p. 40; Trans. A.S.M.E., 7–814, 9–489; Engr. U.S., Oct. 15, 1907.

Domes: Engr. U.S., Jan. 1, 1907, p. 27.

Classification of Boilers and Comparison of Types: Engr. U.S., Jan. 1, 1907; Min. Rept., Feb. 21, 1907.

Furnace and Settings: National Engr., Sept. and Nov., 1907; Elec. World., Sept. 7, 1907; Am. Elecn., Jan., 1902, p. 10, Nov., 1903, p. 557, July, 1904, p. 339; Trans. A.S.M.E., 6–118, 16–590, 19–74, 782, 20–95; Am. Mach., Aug. 18, 1898; Engr. U.S., July 15, 1905, p. 471, Sept. 15, 1905, p. 622, Aug. 1, 1906, p. 491, May 15, 1906, Jan. 1, 1907; Power, March 24, 1908, p. 445, June, 1905; Eng. Mag., July, 1897, p. 587.

Boiler Inspection: Power, Jan., 1906, p. 32; Engr. U.S., Feb. 15, 1907; Trans. A.S.M.E., 4–142; Boiler Maker, Nov., 1907; Cassier's, Feb., 1907.

Riveted Joints: Power, March, 1906, p. 147, April, 1906, p. 227; Engr. U.S., Jan. 1, 1907, p. 21, Aug. 15, 1907, p. 784; Trans. A.S.M.E., 6–120, 10–707; Boiler Maker, June, 1906, Dec., 1907; Prac. Engr., Dec. 13, 1907.

Safety Valves: See paragraph 350.

Specifications: Power, Dec., 1905, p. 728; Nat. Engr., May 15, 1903, p. 367; Eng. News, Oct. 20, 1898, p. 251; Boiler Maker, Sept., 1906, p. 243.

Thickness of Boiler Plate: Am. Mach., Jan. 16 and Feb. 27, 1902; Trans. A.S.M.E., 22–127, 15–629, 24–921; Eng. News, Jan. 31, 1901, p. 121.

Boiler Testing: See A.S.M.E. Code for conducting Standard Boiler Trials, reprinted in Appendix B. See also Power and Engr., Feb. 23, 1909.

Bridge Walls in Theory and Practice: Power and Engr., Mar. 9, 1909, p. 452.

CHAPTER IV.

SMOKE PREVENTION, FURNACES, STOKERS.

87. General. — It is recognized that bituminous coal can be efficiently burned without smoke in a properly designed furnace if proper attention is given to stoking and other factors involved. It is nevertheless a common statement among owners of power plants that it is cheaper to smoke than to operate without smoke. This is undoubtedly true in many cases where smokeless combustion can be secured only by admitting a considerable excess of air with a consequent loss in economy frequently greater than that due to incomplete combustion and smoke. Even under the most favorable conditions, however, smokeless combustion depends largely upon skillful manipulation by an interested and efficient fireman.* The order of intelligence demanded for this work is out of all proportion to the wages paid. In many small plants — and these are usually the most obstinate smoke offenders — the fireman handles as much as a ton of coal per hour by hand, besides caring for the feed pumps and water levels, keeping the boilers clean, and removing the ash. The boiler room is frequently poorly lighted and poorly ventilated. It is, therefore, not surprising that the fireman seldom worries about the smoke problem. A better wage scale and more consideration for the firemen might do a great deal toward abating the smoke nuisance. In order that combustion may be smokeless and efficient, *the volatile gases and separated free carbon must be brought into intimate contact with the proper quantity of air and maintained at a temperature above the ignition point until oxidation is complete before they are brought in contact with the heat-absorbing surfaces of the boiler.* Mere excess of air will not effect smokeless combustion, even if the gases and air are thoroughly mixed, if the temperature is prematurely reduced below that necessary for combustion by contact with the heat-absorbing surfaces of the boiler.

Smoke may be produced, therefore, by

1. An insufficient amount of air for the perfect combustion of the volatile gases. This is primarily a question of draft.

2. An imperfect mixture of air and combustible.

3. A temperature too low to permit complete oxidation of the volatile combustible.

* See Appendix G. Rules for firemen using Illinois and Indiana coal in hand-fired furnaces.

Smoke-preventing devices may be divided into two classes:

(1) Those which are an integral part of the boiler and setting, such as mechanical stokers, Dutch ovens, down-draft furnaces and fire-tile combustion chambers incorporated with the regular setting.

(2) Those which may be conveniently attached to plants already in operation without material modification of the furnace, such as steam jets and other means of mixing air and combustible gases, admission of air through the bridge wall or side wall, and mechanical draft.

88. Mechanical Stokers. — *Uniform evolution of the volatile gases* of the fuel is the essential requisite for smokeless combustion, and it is for this reason that mechanical stokers as a class are more effective in preventing smoke than any apparatus accompanied by intermittent firing. Stokers which feed irregularly have the effect of hand-fired furnaces, and it is necessary not only to employ some powerful auxiliary mixing device but also to furnish at times an extra supply to take care of the enormous volume of volatile gas evolved after a fresh charge of fuel is added.

Carefully adjusted automatic stokers owe their high efficiency to (1) *uniformity of feed;* (2) *proper proportion of air and combustible;* (3) *absence of excessive air dilution, as when the fire doors are opened in connection with hand firing; and* (4) *self-cleaning grates.*

Daily records are essential with any type of stoker or hand firing if efficient results are expected, as only by frequent observation is it possible to determine the proper adjustment of air supply, depth of fire, rate of feed, and the like. Control of air supply is almost as important as the upkeep and effective operation. In the best firing practice the right amount of air, depth of fire, and rate of feed must be worked out by the engineer.

Stokers are often condemned by owners as inefficient and inferior to hand stoking because no particular attention had been paid to them beyond filling the hopper with coal. They should be operated in strict accordance with the principles of design.

In plants of 2000 horse power or over the installation of mechanical stokers and coal conveyors effects a considerable saving of labor and can usually be relied upon to solve the smoke problem if reasonable attention is given to their operation. In smaller plants interest on the investment and other considerations may make hand firing more economical, although many plants of capacities as small as 200 horse power are giving satisfaction, particularly in places where a poor grade of fuel is used and smoke ordinances are rigidly enforced. A stoker of the self-cleaning, slow-running type requires much less attention than the hand-fired furnace. With hand firing one fireman can effi-

ciently attend to the water, coal, and ashes of about 200 horse power or handle coal for say 500 horse power, whereas with good automatic stokers he can readily take care of 2000 horse power or of 4000 horse power with chain grates equipped with overhead bunkers and down spouts.

The best stokers are those which are least complicated and simplest in operation. A cheap stoker is a poor investment, since the cost of repairs and shut downs will usually amount to more than the saving in price.

The following outline gives a classification of a few of the best known American mechanical stokers:

Front Feed.

Chain Grates:
- Babcock & Wilcox,
- Green,
- McKenzie,
- Playford.

Step Grates:
- Roney,
- Wilkinson,
- Acme,
- McClave.

Side Feed.

Step Grates:
- Murphy,
- Detroit,
- Mode.

Under Feed.
- Jones,
- American,
- Taylor,
- Guckett.

Down Draft.
- Hawley

Sprinkler.
- Little Giant,
- Swift,
- Vulcan.

Powdered Fuel.
See paragraphs 34–46.

Mechanical Stokers: Power, March, 1906, p. 189, Aug., 1905, p. 487, March, 1903, p. 112; Jour. West. Soc. Engrs., Feb., 1903, p. 44; Eng. Soc. West Penn., April, 1903, p. 169; Eng. News, March 26, 1903, p. 272; Eng. News, 35–226; Eng. Mag., July, 1902, p. 528; Engr. U.S., Jan. 1, 1907, p. 83, Aug. 15, 1906, p. 440, July 2, 1906, p. 437, Feb. 1, 1904, p. 114, April 1, 1903, p. 262; Elec. Engr., Lond., 33–977; Cassier's, Sept., 1906, p. 469; Trans. A.S.M.E., 17–278, 558; Am. Engr., July, 1905, p. 281, July, 1904, p. 284; Am. Elecn., July, 1904, p. 329, 1902, p. 489, 14–18, 12–411, 263.

89. Chain Grates. — The chain grate, Fig. 61, is one of the most popular forms of automatic stokers. It embodies a moving endless chain of grate bars mounted on a frame with provision for the continuous and uniform feeding of coal into the furnace, the fuel and the grate moving together. The operations of feeding the coal, carrying it through the progressive stages of combustion, removing the ashes and clinkers, and maintaining a clean grate and free air supply are practi-

Fig. 61. Green Chain Grate.

FIG. 62. Babcock and Wilcox Boiler, Chain Grate, Ordinary Setting.

FIG. 63. Babcock and Wilcox Boiler, Chain Grate, Fire-tile Roof.

cally automatic. The driving mechanism consists of a gear train actuated by a cast-steel ratchet and pawls, the arms carrying the latter being given a reciprocating motion by an eccentric mounted on a line shaft. The latter may be driven by any type of engine or motor and the speed of the grate regulated by varying the stroke of the arm carrying the pawls. Fuel is fed into a hopper placed at the front end of the furnace and the depth of the fuel regulated by a guillotine damper. The entire grate and driving mechanism are mounted on a permanent

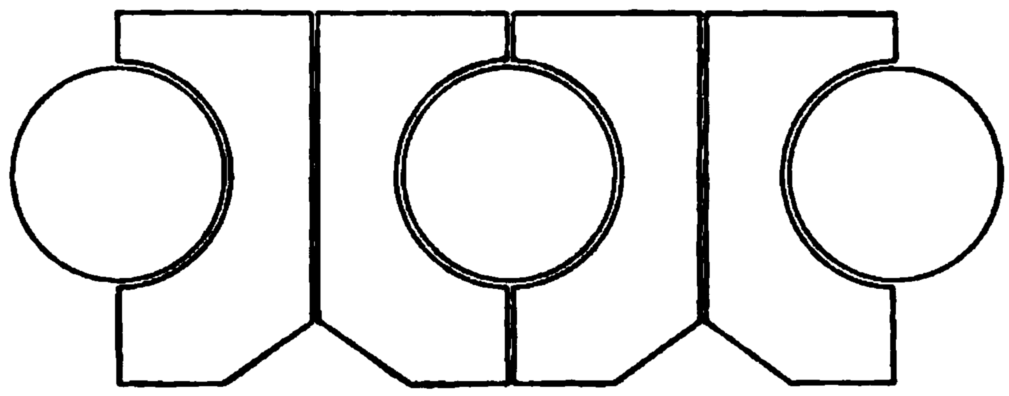


FIG. 64. Section of Tiles Encircling Lower Row of Tubes.

truck and may readily be removed from beneath the boiler. The front part of the furnace is provided with a flat or slightly inclined ignition arch as illustrated. The thickness of the fire and the speed of the grate should be so regulated that when the fuel has reached the end of the grate it shall have been completely consumed and ashes only will be discharged into the pit. The combination of chain grate with inclined igni-

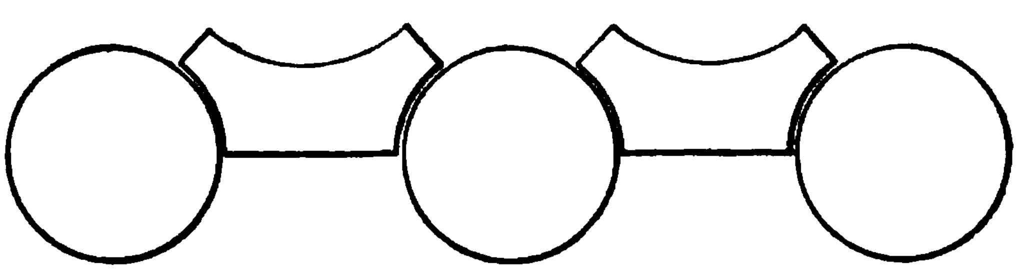


FIG. 65. Tiles Between Lower Row of Tubes, Back of Encircling Tiles.

tion arch, curved bridge wall, and lower course of tubes covered with fire tiling, as shown in Fig. 61, makes an excellent smokeless furnace, though the depth of fire and air supply must be carefully regulated to prevent excessive air dilution. With chain-grate stokers there may be a considerable leakage of air between the grate and bridge wall, through the coal in the hoppers, under the coal gate, and through the fire bed at the rear where it is mostly ash. Various schemes have been employed to prevent leakage at the end of the grate by using water backs, ash-pit dampers, and the like. Fig. 62 shows the application of the chain grate

to a Babcock & Wilcox boiler with the standard arrangement of fire tiles and bridge wall. To insure the best coking effect with this arrangement the flat ignition arch is employed. This reduces the velocity and increases the temperature of the gases as they are distilled from the green fuel, thereby assisting complete combustion. Under normal conditions of operation this insures practically smokeless combustion, but is not always successful at heavy overloads, since the length of furnace is not always great enough to thoroughly mix the air and combustible gases before they reach the boiler surfaces. A more satisfactory arrangement is the modification illustrated in Fig. 63. The coking arch is slightly inclined and the gases are compelled by means of the fire-tile roof to take a longer path to the rear before crossing the tubes to the uptake, which must necessarily be at the front end of the boiler. Tests have shown this arrangement to give excellent efficiency of boiler and furnace, with practically smokeless combustion, though the cost of the upkeep of the tiling is very high.

Figs. 64 and 65 illustrate the method employed in suspending the fire tiles. Another method of increasing the extent of regenerative fire-brick surface without disturbing the standard setting is to cover the tubes over the furnace with a fire tile, which permits the gases to flow between the tubes. Such an application to a Stirling boiler is illustrated in Figs. 66 and 67. The useful life of this class of tiling is very short and this fact prevents its adoption in most cases.

Fig. 67a and Fig. 67b show recent applications of chain grates to large B. & W. boilers which are effecting high efficiency and heavy overload capacity with practically smokeless combustion. See *Power* and *The Engineer*, May 31, 1910, p. 981.

Chain grates properly installed and taken care of cost very little for maintenance. At the South Side Elevated Railroad Station, Chicago, where there are 22 B. & W. chain grates, the total cost of repairs for the grates alone is stated to have been but $25 in eight years. Flat fire-brick arches as in Fig. 62 have to be renewed about once a year. The inclined arch with a good quality of fire brick should last two years or more. The cost of renewing arches of this type approximates $200 for 500-horse-power boilers per setting. The fire-tile furnace is more costly in yearly repairs than the common furnace, but the increased efficiency may offset the extra cost. (A. Bement, *Eng. U. S.*, June 1, 1907, p. 606.) The repairs for a 500-horse-power Babcock & Wilcox boiler setting as in Fig. 63 is approximately $350 per year for arches and tiling.

90. Step Grates, Front Feed. — Fig. 68 shows the general arrangement of a Roney stoker and Fig. 69 that of a Wilkinson stoker, illus-

FIG. 66. Application of "Economy" Fire Tiles to Stirling Boiler.

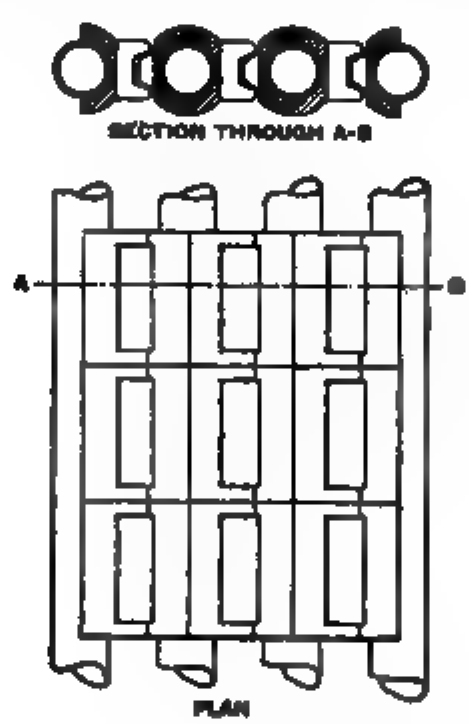


FIG. 67. Method of Anchoring "Economy" Fire Tiles to Tubes.

Fig. 67a. Chain Grate Fired from Rear End of Setting as Installed at Quarry Street Station, Commonwealth Edison Co., Chicago.

FIG. 57b. Smokeless Setting, Boiler Units 5 and 6, Quarry Street Station, Commonwealth Edison Co., Chicago.

trating the step-grate, front-feed principle. The Roney stoker consists of a hopper for receiving the coal, a set of rocking stepped grates inclined at a proper angle from the horizontal, and a dumping grate at the bottom of the incline for receiving and discharging the ash and clinkers. The dumping grate is divided into several sections for convenience in handling. The coal is fed on to the inclined grate from the hopper by a reciprocating "pusher" actuated by the "agitator." The power is supplied through an eccentric operated by a small engine or motor. The normal feed is about 10 strokes per minute. The grate bars rock

Details of Construction of the Roney Mechanical Stoker

FIG. 68. Details of Roney Stoker.

through an arc of 30 degrees, assuming alternately horizontal and inclined positions. The construction permits abundance of air to pass through the fuel, with little or no possibility of coal dropping through the grate. A coking arch of fire brick is sprung across the furnace as indicated. This stoker operates with natural draft and with suitable arrangement of fire tiling effects complete and efficient combustion. Without a fire tile roof construction smokeless combustion is effected with difficulty, particularly at heavy loads.

In the Wilkinson stoker the inclined grate bars are hollow and are arranged side by side, every alternate bar being movable. When in

FIG. 68a. Double Stoker Installation at the 59th Street Station of the Interborough Rapid Transit Co., N. Y.

operation there is a constant sawing action of the grate bars, causing the fuel to flow forward and downward. A small steam jet with about $\frac{1}{16}$ inch opening is introduced into the end of each hollow grate bar, thus inducing the required amount of air for combustion, which passes through air openings approximately $\frac{1}{4}$ inch wide by 3 inches

Fig. 69. Details of Wilkinson Stoker.

long. These stokers are driven by two small hydraulic motors, the water being furnished by a small pump and being used over and over again.

91. Step Grates, Side Feed. — Fig. 70 shows a front vertical section and Fig. 71 a side vertical section through a Murphy automatic stoker and furnace. The apparatus is in effect a Dutch oven equipped with an automatic feeding and stoking device. Coal is introduced either mechanically or by hand into the magazine at each side of the furnace and above the grate and descends by gravity upon the coking plate. Reciprocating stoker boxes push the coal out upon the grate bars. Every alternate grate bar is movable and pivoted at its upper end. A rocker bar driven by a small motor or engine causes the lower ends to move up and down, this action producing the required stoking effect. A device for grinding up the clinker and ash is provided as shown at the bottom of the furnace. This is hollow and is connected by a 2-inch

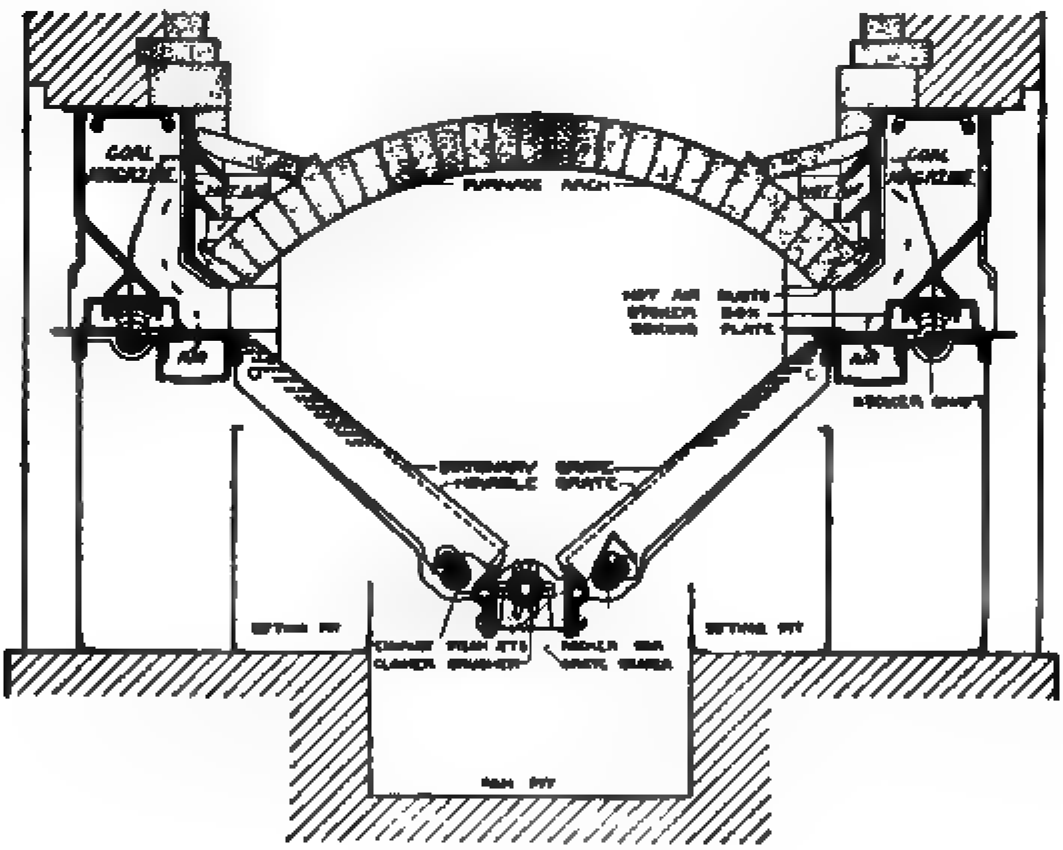


FIG. 70. Murphy Furnace, Front Section.

FIG. 71. Murphy Furnace, Side Section.

pipe with the smoke flue, so that the cold air passing through it prevents it being destroyed by the heat. Air is supplied to the green coal through flues passing under the coking plates, and the speed of the stoker boxes and grate bars can be regulated to conform to any rate of combustion. On account of the large fire-brick combustion chamber, this stoker with *careful manipulation* is capable of practically smokeless combustion. The power house of the Northwestern Elevated Railroad Company, Chicago, Ill., is equipped with Murphy furnaces, which are operating smokelessly at an unusually high combustion rate, whereas a number of other installations using the same type of stoker and boiler and burning the same class of fuel are heavy smoke producers. Murphy furnaces are peculiarly adapted to variable loads, since at light loads the stoker may be operated with reduced grate area by allowing the bottom of the grate to partly fill with ashes.

92. Underfeed Stokers. — Fig. 72 shows the general principles of the Jones underfeed stoker. It consists of a steam-actuated ram with a

Fig. 72. Jones Underfeed Stoker.

fuel hopper outside the furnace proper and a retort or fuel magazine and auxiliary ram, *A*, *A*, within. Heavy cast-iron tuyère blocks for the admission of air are placed on either side of the retort. Fuel is forced underneath the fire by the ram and its auxiliary, the ram movement of the fuel being backward and upward, displacing the incandescent fuel and ash and forcing it on to the dead plates. There is no ash pit, the ashes being raked from the dead plate by hand. Air, supplied by a blower, is admitted through the openings in the tuyère blocks. The latter are at a point above the green fuel in the retort tube, but below the fire. The standard size of retort is about 6 feet in length by 24 inches in width and 18 inches in depth, and experience has shown that other sizes are not necessary, since the spaces between retort and side walls of the various furnaces may be provided for by extending the width of dead plate. One stoker is usually installed for each furnace, though two are sometimes required. The draft from the fan and the number of strokes of the ram are automatically con-

trolled by the steam pressure, although provision is made for regulating either by hand. Underfeed stokers are adaptable to all grades and sizes of coal and on account of the forced draft are capable of burning very low grades of fuel. A number of these stokers are installed in the power plant of the First National Bank Building, Chicago, and are giving high efficiency and smokeless combustion with low-grade Illinois screenings. The cost of upkeep is rather high when compared with chain grates.

Fig. 73 shows an application of an American underfeed stoker to a return tubular boiler. This differs from the Jones stoker in the method

FIG. 73. American Underfeed Stoker.

of feeding the fuel to the retort and in the employment of "live" grates instead of dead plates on the sides of the retort. The coal is fed into the hopper and carried by an endless screw conveyor into the magazine or retort. Forced draft is used and the rate of draft and the speed of the conveyor are readily adjusted to suit the conditions of load. Underfeed stokers are very compact, occupy but little space in front of the boiler, and are low in first cost. Careful manipulation is necessary to render them smokeless and efficient, since they are ordinarily installed without fire-tiled combustion chambers.

93. Down-Draft Furnaces. — Fig. 74 shows the application of a Hawley down-draft furnace to a Heine water-tube boiler. In this furnace there are two separate grates, one above the other, the upper

FIG. 74. Hawley Down-Draft Furnace Applied to a Heine Boiler and Equipped with Fire-Tile Combustion Chamber.

one being formed of parallel water tubes connected with the water space of the boiler through the steel headers or drums *A* and *D*, in such a manner as to insure a positive circulation. Fuel is supplied to the upper grate, the lower one, formed of common bars, being fed by the half-consumed fuel falling from the upper grate. Air for combustion enters the upper fire door, which is kept open, and passes first through the bed of green fuel on the upper grate and then over the incandescent fuel on the lower grate. A strong draft is required, due to the relatively small upper-grate area and the correspondingly high rate of combustion. The down draft is very well adapted to the burning of paper, cardboard, excelsior, wood, and other rapidly burning refuse, as well as the various kinds of coal. Lump coal gives better results than the smaller sizes, as the latter are apt to fall through the upper grate before even partially consumed and when such is the case efficient results cannot be obtained. If carefully manipulated this furnace with fire-tiled tubes as illustrated in Fig. 74 gives high boiler efficiency and smokeless combustion,

but its overload capacity is limited. Without the fire tiling smokeless combustion is possible only at light loads. Hawley down-draft furnaces are installed in a large number of tall office buildings in New York and Chicago and are giving excellent results. Down-draft furnaces are necessarily hand fired, since mechanical stokers are not easily adapted to them.

94. Sprinkling Stokers. — In this system of stoking the fuel in finely divided form is distributed by sprinkling uniformly over the entire area of the grate. With the proper adjustment of air supply and feed the volatile gases are distilled off continuously before the grate is covered by the new coal and without materially lowering the temperature of the incandescent fuel bed. Mechanically the operation involves considerable difficulty. One of the most successful American devices for this purpose is the "Little Giant" stoker. Coal of nut size or smaller is hand fed into a small hopper from which it gravitates on to a feed wheel driven by an engine or motor. A stream of coal is discharged into a cast-iron chute extending over the front part of the grate, from which it is blown into the furnace by a steam jet. The fine or powdered coal is burned in suspension and the heavier coal falls to the grates. A fire-brick combustion chamber is usually necessary for smokeless combustion, since the quantity of fuel burned may be too large to permit complete distillation of the green fuel before the succeeding charges are delivered. A test on an English stoker of this type (Bennis stoker), using compressed air for feeding the fuel, gave the unusually high rate of combustion of 72 pounds of coal per square foot of grate surface per hour. The same test credited the boiler and grate with an efficiency of 84.9 per cent, which is probably the highest recorded efficiency of any boiler and furnace using coal as fuel. (*Engineering Record,* April 8, 1905, p. 404.)

95. Dutch Ovens. — An independent furnace or Dutch oven in front of the boiler as illustrated in Fig. 75 provides one of the simplest methods of securing a large combustion chamber for the mingling of the air and combustible gases before delivering them to the boiler proper. Such a furnace produces very high temperatures when operating under best conditions, and hence must be lined with fire brick of excellent quality. Although better than the ordinary setting the plain Dutch oven is too limited in length and capacity to prevent smoke from forming, except at very light loads. The velocity of the gases is usually too high to permit either a thorough mixture or complete oxidation before striking the boiler tubes. Steam jets placed at the sides of the setting and blowing across the fire are sometimes effective in mixing the air and combustible gases, but the best results are obtained by modifying the construction of the furnace to the extent of introducing

baffle walls which vary the direction of flow and by increasing the length of the path of the heated gases. The greater the length of the path and the greater the number of baffles the more thoroughly will the air and gases be mingled, but the intensity of draft will of course be decreased in proportion. A compromise must therefore be made between required draft and length of path. The larger the extent of fire-tile surface the greater will be the regenerative effect, which is of particular importance in hand firing when the evolution of volatile gas is intermittent, but the first investment and cost of repairs and renewals are greater. There are little reliable data available pertaining to the relation between capacity of furnace and length of path of the heated gases for maximum

FIG. 75 Plain Dutch Oven.

efficiency. A modified Dutch oven is illustrated in Fig. 71. The extension front is not necessary with some type of boilers, as will be seen from Figs. 63 and 74, in which a tile roof and baffles suitably arranged within the setting proper simulate the Dutch oven effect. (See "Cost of Maintenance of Dutch Oven Furnaces," A. Bement, *Eng., U. S.*, July 1, 1907, p. 606.)

96. Twin-Fire Furnace. — This arrangement, illustrated in Fig. 76 in connection with a hand-fired return tubular boiler, is a double furnace formed by longitudinal arches extending between bridge wall and fire door.

The furnaces are fed and manipulated alternately, the object being to have one furnace in a highly incandescent state while green fuel is fed into the other. Air is admitted both below and above the grate, and the volatile gases are supplied with the necessary oxygen for com-

bustion before they come in contact with the comparatively cool boiler surface.

The gases from both furnaces first pass into a chamber formed by a single arch sprung across the entire inner setting from the side wall, a short retarding arch being placed between this intermediate chamber and the rear of the setting. A special tile of high-grade refractory clay is used, the thickness varying from 4 to 6 inches, depending upon the size of furnace and the length of span. The furnace can readily be substituted for the ordinary types in common use under any standard tubular or water-tube boiler and may be installed either under the boiler, as indicated in the illustration, or in an extension Dutch oven. This is an excellent furnace, and when properly manipulated gives smokeless and efficient combustion.

FIG. 76. "Twin Fire Arch" Applied to a Return Tubular Boiler.

96a. Chicago Settings for Hand-Fired Return Tubular Boilers. — Figs. 67a, 67b and 67c show the general details of settings for return tubular boilers as recommended by the Chicago Department of Smoke Inspection. The setting illustrated in Fig. 67a is ordinarily installed where a strong draft is available and that shown in Fig. 67b or 67c where the draft conditions are not favorable. All three settings require careful manipulation for smokeless combustion as is the case with hand-

Fig. 76a. Smokeless Setting for Hand-Fired Return Tubular Boiler with Single Arch Bridge-Wall.

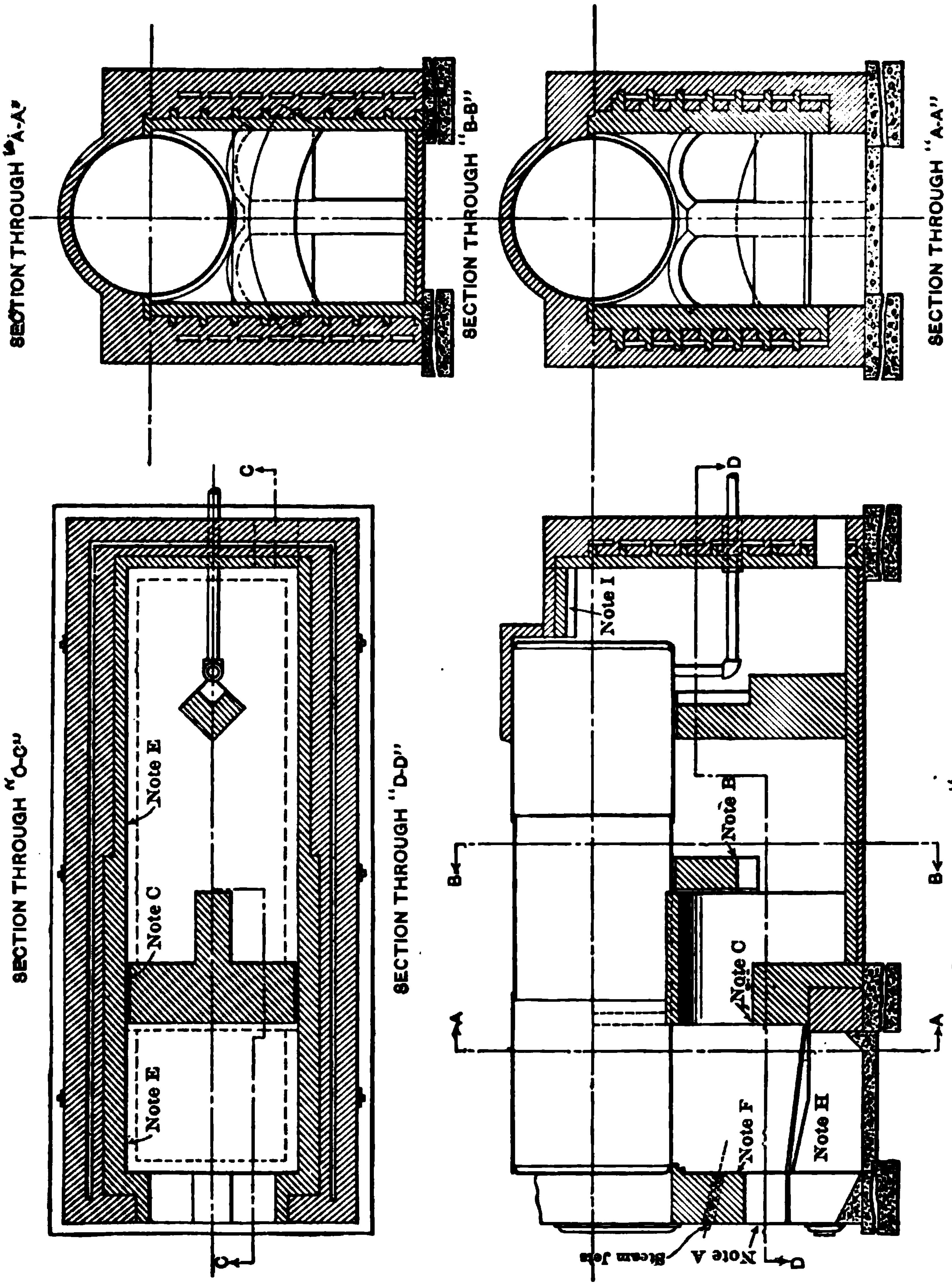


FIG. 76b. Smokeless Setting for Hand-Fired Return Tubular Boiler with Double Arch Bridge-Wall.

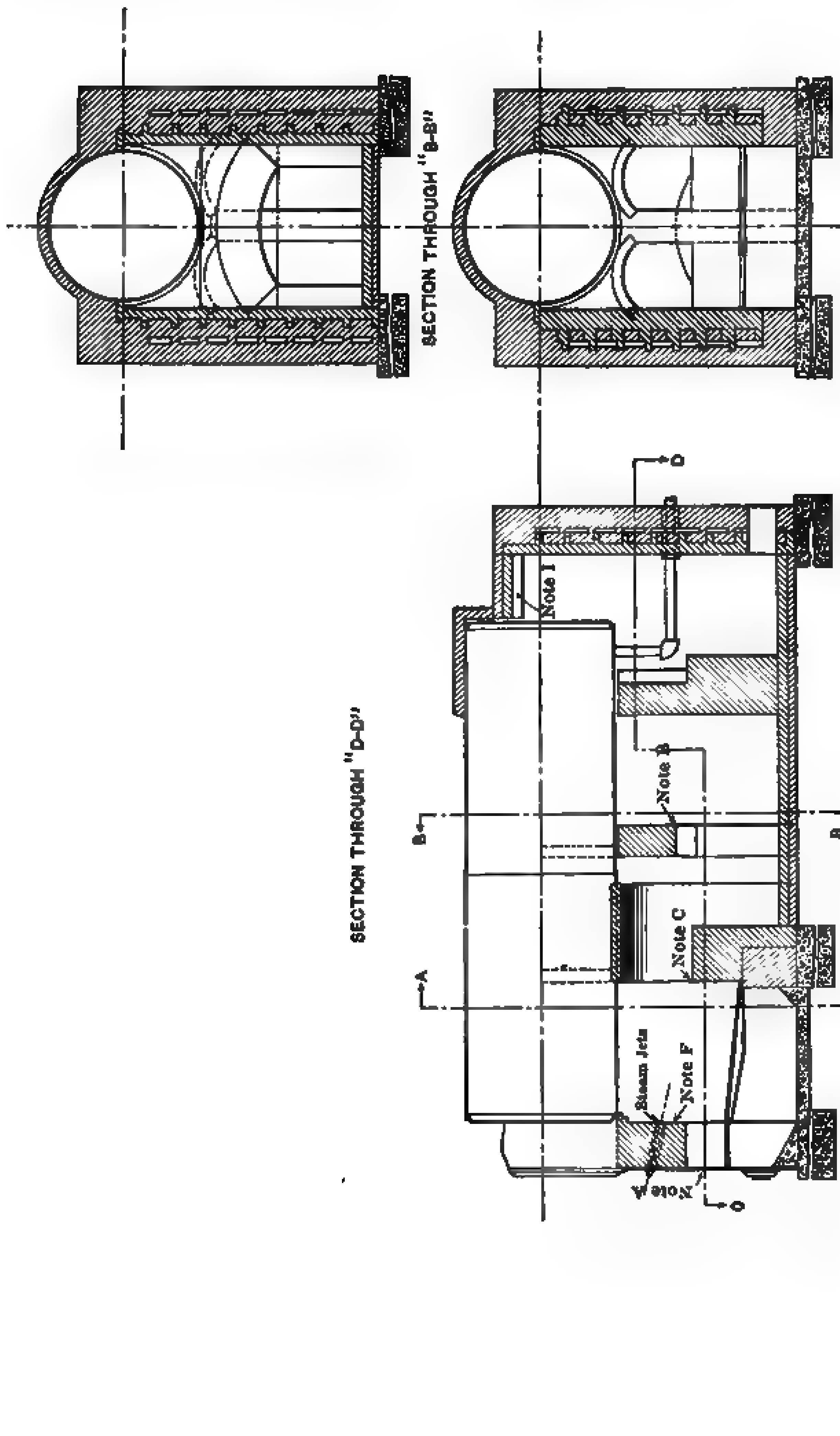


FIG. 76c. Smokeless Setting for Hand-Fired Return Tubular Boiler with Double Arch Bridge-Wall. (Ring Seam Exposed for Inspection.)

fired furnaces in general. It has been the experience of the Department that most violations of the smoke ordinance are due primarily to insufficient draft, the required rate of combustion being too high for the available air supply. The requirements outlined in paragraph 95 apply equally well to these settings. The following specifications refer to Figs. 67a, 67b and 67c, the items in the specifications corresponding to the letters in the illustrations.

A. Doors should be of a type allowing the admission of excess air over the fire when so desired. If panels are cut in the fire doors for this purpose, the aggregate area of the openings should be not less than 4 square inches to each square foot of grate surface.

B. Arches should be made of wedge brick or "bull heads" and not laid in two courses of 4½-inch brick.

C. The bridge wall should be made of first grade fire brick above the grate line and with fire brick facing not less than 9 inches in thickness on the combustion chamber side. The top row should be a row-lock course. Provision should be made in the building of the bridge wall for lateral expansion.

D. The combustion chamber floor should be paved with fire brick laid on edge.

E. Fire brick lining below the arch skew-backs should be not less than 9 inches in thickness. Fire brick lining above the arch system and behind the deflection arch may be 4½ inches in first grade fire brick, with headers every fifth row.

F. Fire brick over firing door liners should be arched. This rule also applies to brick above the clean-out door openings.

G. Facilities for taking up arch thrust should be provided in every case by suitable metal re-enforcements extending horizontally throughout the length of the arches. No air space should intervene between the metal re-enforcement and the skew-backs.

H. Herringbone or Tupper grates or other similar types should not be selected where bituminous coal forms the major portion of the fuel.

I. The back arch is preferably sprung from side to side rather than from back wall to rear boiler tube sheet. No metal should be exposed to direct heat of gases.

J. Chimney heights of less than 75 feet above the grate line should not be permitted, and this height allowed only when the chimney is direct connected to the boiler uptake. In case of a breeching and detached chimney, add to the height of chimney computed by standard methods (never less than 75 feet) 10 feet for every turn of the breeching and one foot for each foot in length of the breeching.

K. For boilers 48 inches or less in diameter, special provision for the examination of girth seams must sometimes be made. This is because of the fact that with small boilers there is not sufficient room between the arch and shell for purposes of inspection.

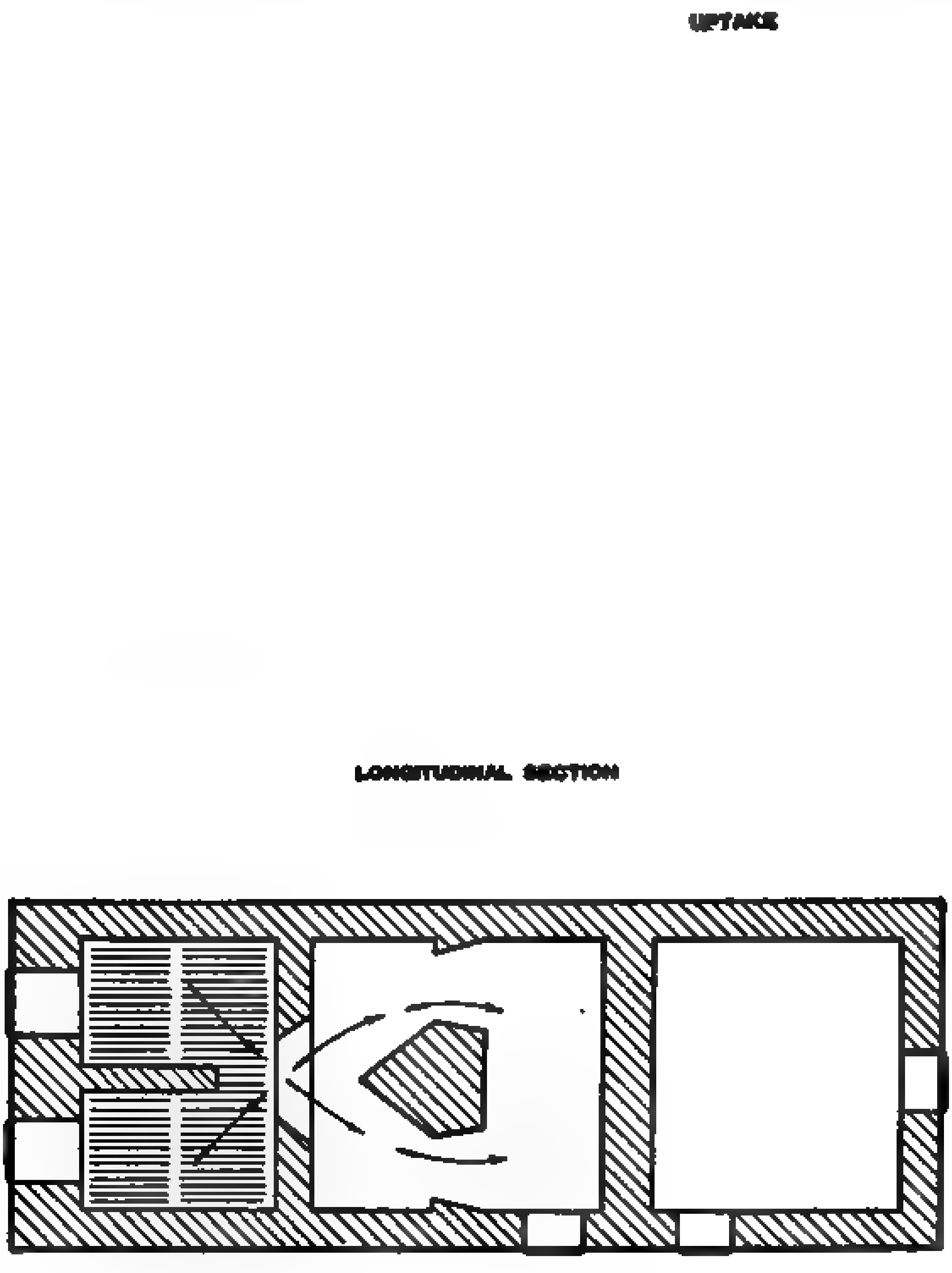


Fig. 77. Wooley Smokeless Furnace.

L. In the event of arch failures, the boiler should be immediately taken out of service. This is to avoid failure of the boiler shell due to heat being applied upon a portion of the heating surface over which a mud deposit has formed.

97. Wooley Smokeless Furnace. — Fig. 77 shows a longitudinal section and a sectional plan of a Wooley smokeless furnace applied to a B. & W. boiler. The main features of the furnace are a dividing wall in the fire box and a deflecting wall in the combustion chamber. The dividing wall permits of the alternate method of firing, whereby one side of the furnace is always in an incandescent state while the other side is being supplied with green fuel. If a mechanical stoker is used the wall in the fire box is omitted. The products of combustion are intended to be thoroughly mingled with the requisite amount of air by the deflecting walls before entering the regenerative or secondary combustion chamber.

Fig. 78. Kent's Wing Wall Furnace.

98. Kent's Wing-Wall Furnace. — Fig. 78 shows the application of Kent's wing-wall furnace to a water-tube boiler. The Dutch oven in front of the regular setting contains the grates. Wing walls *F* are placed as shown two or three feet to the rear of the bridge wall *D*, and fire-brick piers *H* behind the wing walls.

In operation, fresh coal is spread alternately over each half of the grate. The dense smoky gases which rise from the green portion of the fire mingle in the narrow passage with the highly heated air which comes through the other side of the grate greatly in excess of that required to consume the partially burned coal there. The piers *H* act

as regenerative surfaces, absorbing heat from the fire when it is hottest and giving it out when it is coolest, that is, just after firing.

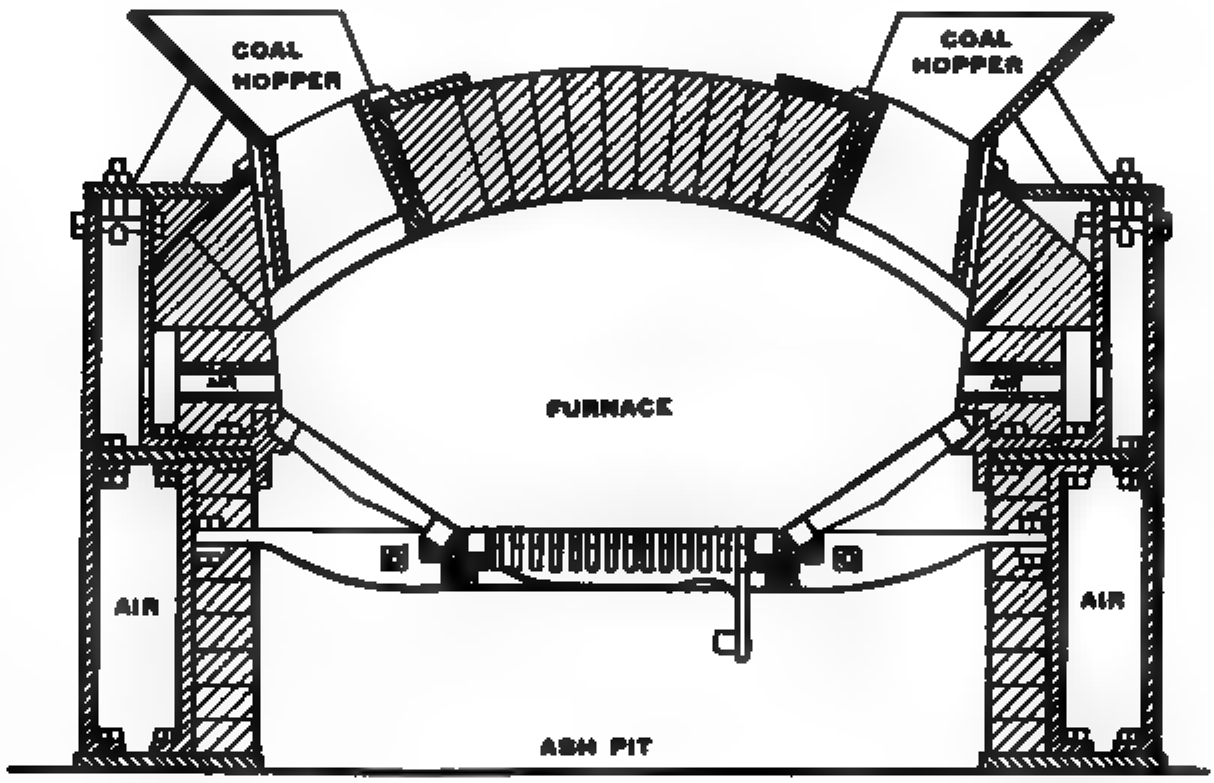


FIG. 79. Burke's Smokeless Furnace, Front Section.

FIG. 80. Burke's Smokeless Furnace, Side Section.

Comparative tests of boilers with standard setting and with wing-wall furnaces have shown a much higher efficiency with the latter and with practically smokeless combustion. (*Iron Trade Review*, July 7, 1904, p. 76.)

99. Burke's Smokeless Furnace. — Figs. 79 and 80 show sections through a Burke smokeless furnace as installed in a number of tall office buildings in Chicago. It amounts virtually to a Dutch oven equipped with shaking grates, and embodies an extension self-feeding coking oven of cast-iron section lined with fire brick and protected from overheating by air circulation through the sections. Natural draft is used, the fire doors being closed; but air is admitted above as well as below the fire. As this stoker is manipulated by hand, more or less attention is required of the operator in keeping the fire clean. Furnaces of this type at the power plant of the Majestic Theatre building, Chicago, Ill., are giving excellent results.

100. Admission of Air above Fire. — Smoke is often due to insufficient air supply or imperfect mixing, especially when coal of a coking or clinkering nature is used which tends to seal up the air spaces in the grate. In these cases the admission of air above the grate through openings in the bridge wall or passages in the side walls frequently gives satisfactory results. When natural draft is not sufficient, as is usually the case under heavy load, steam jets or forced draft may be employed. For a description of such devices see paragraphs 148 to 150.

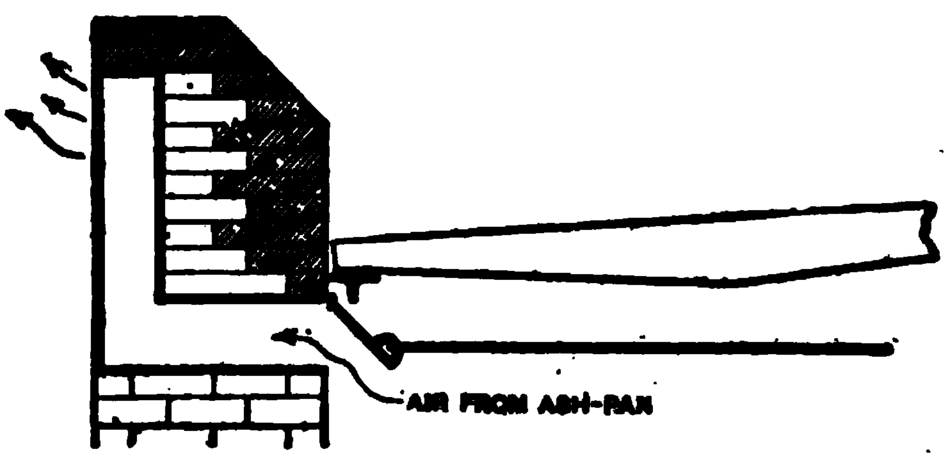


Fig. 81. Split Bridge Wall.

101. Cost of Stokers.—The following is the approximate cost of stokers suitable for a Babcock & Wilcox boiler of 350 horse power rated capacity with 45 square feet of grate surface; height of chimney above grate, 175 feet; coal burned, Illinois screenings. The cost of installation is not included.

1. Chain grate and appurtenances	$1,500.00
2. Jones underfeed stoker	1,400.00
3. Hawley down-draft furnace	1,350.00
4. Burke smokeless furnace	1,000.00
5. Roney stoker	1,300.00
6. Murphy furnace and stoker	1,350.00
7. Wilkinson stoker	1,200.00

Parsons' Smokeless Furnace. See par. 149.
Heinrich Smokeless Furnace. See par. 150.
Steam jets. See par. 148.
Hamler-Eddy Smoke Recorder. See par. 411a.
Ringlemann Smoke Chart. See p. 765.

Smoke Prevention: Bulletin No. 15, Univ. of Ill., Vol. 431; Bulletin No. 334, U. S. Geological Survey; Boiler Maker, May, 1909, Oct., 1909; Cassier's Mag., Feb., 1907; Minn. Engr., Jan., 1910.

Mechanical Stokers: Engr. U.S., Jan. 1, 1907, p. 83, Aug. 15, 1906, p. 540, July 2, 1906, p. 437; Cassier's Mag., Sept., 1906, p. 469; Power, Mar., 1906, p. 189, Aug., 1905, p. 487.

CHAPTER V.

SUPERHEATED STEAM; SUPERHEATERS.

102. General. — The steam engine fails to realize the efficiency of the ideal engine chiefly on account of cylinder condensation. The loss in heat due to this cause is seldom less than 10 per cent of the total supplied, and often as great as 40 per cent.

If the steam is superheated before being admitted to the cylinder, condensation may be reduced or prevented entirely, as was recognized as early as sixty years ago, but the mechanical difficulties encountered prevented the practice until within the past few years.

The principal advantages of superheated steam in connection with steam-engine work are:

1. At high temperatures it behaves like a gas and is therefore in a far more stable condition than in the saturated form. Considerable heat may be abstracted without producing liquefaction, whereas the slightest absorption of heat from saturated steam results in condensation. If superheat is high enough to supply not only the heat absorbed by the cylinder walls but also the heat equivalent of the work done during expansion, then the steam will be dry and saturated at release. This is the condition of maximum efficiency in a single cylinder. (Ripper, "Steam Engine Theory," p. 155.) Greater superheat than this will result in a loss of energy unless the steam is exhausted into another cylinder. To obtain dry steam at release the steam at cut off must be superheated 100 to 300 degrees F. above saturation temperature, depending upon the initial condition of the steam and the number of expansions, a higher degree of superheat being required for earlier cut off. A superheat of 200 to 275 degrees F. at admission is necessary to insure dry steam at release in the average single-cylinder engine cutting off at one-fourth stroke, boiler pressure 100 pounds gauge. In most cases superheat is only carried so far as to reduce initial condensation, the steam becoming saturated at cut off, thus permitting efficient lubrication. There will be a reduction of approximately 1 per cent in cylinder condensation for every 7.5 to 10 degrees of superheat. In compound and triple-expansion engines the steam is ordinarily superheated between each stage as well as before admission to the high-pressure cylinder.

2. A moderate amount of superheat produces a large increase in volume, the pressure remaining constant, and diminishes the weight of steam per stroke for a given amount of work. For example, the volume of 1 pound of saturated steam at 150 pounds pressure (gauge) is 2.75 cubic feet, and its temperature is 365.8 degrees F. The total heat of one pound of this steam above the freezing point is 1193.5 B.T.U. By adding 110 B.T.U. in the form of superheat its temperature will be increased to 565.8 degrees F. (superheated 200 degrees F.) and its volume to approximately 3.5 cubic feet (specific heat taken as 0.55).* Thus an increase of 9.2 per cent in the heat effects an increase of 22 per cent in the volume, which means a corresponding reduction in the steam admitted to the engine per stroke. These figures are purely theoretical, as no allowances have been made for condensation of the saturated steam or for reduction in temperature of the superheated steam.

3. Superheated steam has a much lower thermal conductivity than saturated steam, and therefore, less heat is absorbed per unit of time by the cylinder walls.

General Discussion of Superheated Steam: Engr., Lond., Dec. 31, 1909; Eng., Lond., Sept. 13, 1901, Sept. 4, 1903, p. 237; Eng., U.S., Dec. 15, 1902, p. 821, Oct. 15, 1906, p. 687; Engr. Mag., Feb. 1903, p. 778, Sept., 1903, p. 897, Feb., 1904, p. 757, June, 1904, p. 436, March, 1905, p. 943, Nov., 1905, p. 271, May, 1906, p. 269; Eng. Rec., July 8, 1905, p. 28; June 30, 1906, p. 783, July 28, 1906, p. 86; Power, Aug., 1904, p. 463, Sept., 1904, p. 558, Oct., 1904, p. 762, Jan., 1905, p. 23, Feb., 18, 1908, Serial; Eng., Lond., Jan. 8, 1904, p. 42; West. Elec., Nov. 14, 1903, p. 369; Proc. A.S.M.E., May 14, 1908.

103. Economy of Superheat. — Many comparative tests of engines using saturated and superheated steam under varying conditions of pressure and temperature have been made during the past few years, showing in most cases a gain in favor of superheat due to the reduction in steam consumption, but in some cases the extra investment and cost of maintenance neutralize this gain, resulting in an actual loss when measured in dollars and cents per horse-power hour.

As far as steam consumption per horse-power hour is concerned, superheating usually increases the economy five to fifteen per cent

* The most satisfactory equation for determining the specific volume of superheated steam is that given by Knoblauch, Linde, and Klebe (Peabody, "Steam and Entropy Tables," p. 22):

$$pv = 0.5962\,T - p\,(1 + 0.0014\,p)\left(\frac{150{,}300{,}000}{T^3} - 0.0833\right).$$

p = pressure, pounds per square inch absolute.
T = absolute temperature of the steam, degrees F.
v = specific volume of superheated steam, cubic feet.

and in some instances as much as forty, the latter figure referring to the more wasteful types of engines. A fair estimate of the average reduction in steam consumption per horse-power hour with moderate superheating, that is, 100 to 125 degrees F., based on continuous operation of existing plants, is:

	Per Cent.
1. Slow running, full stroke, or throttling engines, including direct acting pumps	40
2. Simple engines, non-condensing, with medium piston speed, including compound direct acting pumps	20
3. Compound condensing Corliss engines	10
4. Triple-expansion engines	6

A prominent European builder of engines guarantees steam consumption with highly superheated steam as follows:

	Pounds per I.H.P. hour.
Single-cylinder condensing engines	13.5
Single-cylinder non-condensing engines	15.5
Compound condensing engines	10
Triple-expansion condensing engines	8.75

In comparing the performances of engines using saturated and superheated steam it is advisable to base all results on the *heat* consumed per horse power rather than on the steam consumption, since the latter is apt to give a false idea of the relative economies. The real measure of economy is the *cost* of producing power, taking into consideration all charges, fixed and operating, and the next best is the coal consumption per I.H.P. hour, but as a means of comparing the engines only, the heat consumption per horse power per hour or per minute is very satisfactory.

See paragraph 181 for the influence of superheat on the economy of reciprocating engines and paragraph 193 for the influence on steam turbines.

Economy of Superheat: Eng. Mag., Dec., 1904, p. 757, April, 1905, Sept., 1903, p. 108; Trans. A.S.M.E., 22–899; Engr. Rec., July 8, 1905, p. 28; Power, Sept., 1904, p. 558, Oct., 1904, p. 598, Jan., 1905, p. 23; Cassier's, Nov., 1903, p. 18.

104. Limit of Superheat. — In this country steam temperatures exceeding 500 degrees F. are seldom employed, while in Europe few if any plants are installed without superheaters, and 600 degrees F. is a common temperature.

Experience has shown that with engines of ordinary design, slide-valves and Corliss, the temperature at the throttle should not exceed 500 degrees F. This corresponds to a superheat of 160 degrees F. with

steam at 100 pounds gauge pressure, and 130 degrees F. at 150 pounds. This degree of superheat insures practically dry steam at cut off in the better grade of engines. Just how far superheating can be carried with a given engine of ordinary construction can be determined by experiment only, but a temperature of 500 degrees F. is probably an outside figure and 450 degrees F. a good average. Higher temperatures are apt to interfere with lubrication and sometimes cause warping of the valves. With temperatures below 450 degrees F. no difficulties are ordinarily met with. Metallic packing has been found to give the best results for both piston rods and valve stem.

It is generally assumed that a greater quantity of oil is required for lubricating valves and cylinders in connection with superheated steam, but experience seems to show that such is not the case. (Proc. A.S.M.E., May 14, 1908.) Forced-feed lubricators are the most satisfactory for superheated steam engines, since they insure a positive and copious flow of oil directly to the valves or other parts requiring it.*

With highly superheated steam involving temperatures of 600 degrees F. or more the poppet-valve type of engine is ordinarily employed, though balanced piston valves are not uncommon. The poppet valve is not distorted by heat and requires no lubrication. In Europe these engines have been brought to a high state of efficiency, but have not been generally adopted in this country owing, no doubt, to the higher cost.

105. Specific Heat of Superheated Steam.† — The total heat of superheated steam is given as

$$H = \lambda + C_p t, \tag{17}$$

in which

λ = B.T.U. in one pound of saturated steam above 32 degrees F.

C_p = mean specific heat of the superheated steam at constant pressure.

t = degree of superheat, degrees F.

Regnault determined the mean specific heat at atmospheric pressure to be 0.48 between 127 degrees and 226 degrees C. of superheating, and until recently this has been assumed to apply to all pressures and temperatures. As early as 1876 Hirn concluded from experiments made with a throttling calorimeter that the specific heat of saturated steam increased with the pressures and decreased at any given pressure if the steam became superheated. Since then numerous investigators have promulgated theories pertaining to this subject which have been far from harmonious and none has been universally accepted. Some experiments appear to show that specific heat is independent of pressure

* Effect of Superheated Steam on Cylinder Oils. Mech. Engr., Lond., July 31, 1908, p. 115.

† See paragraph 113a.

and degree of superheat, while others indicate an increasing value as the pressure and degree of heat increase. Still others corroborate Hirn's theory.

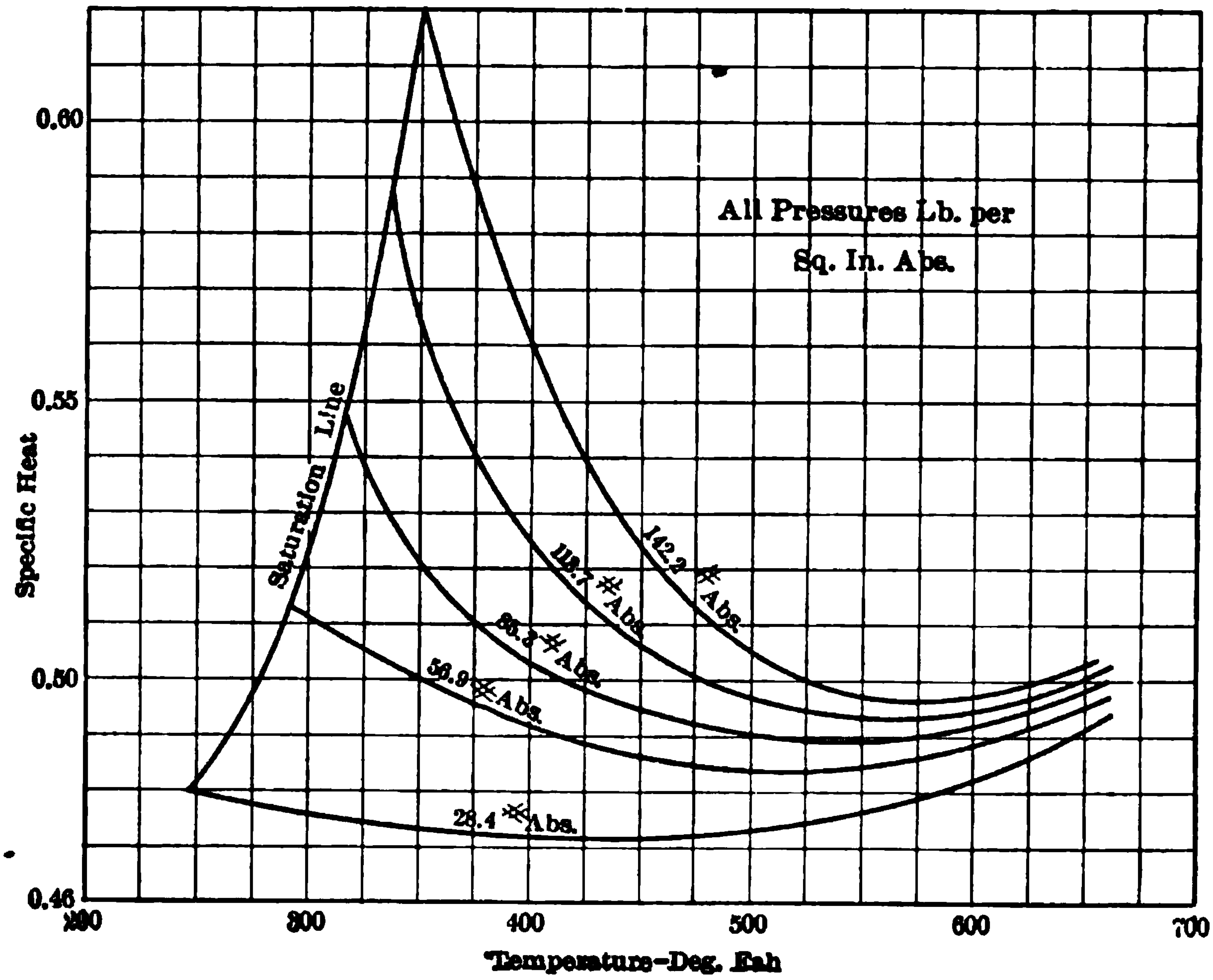


FIG. 82. Specific Heat of Superheated Steam, Knoblauch and Jakob.

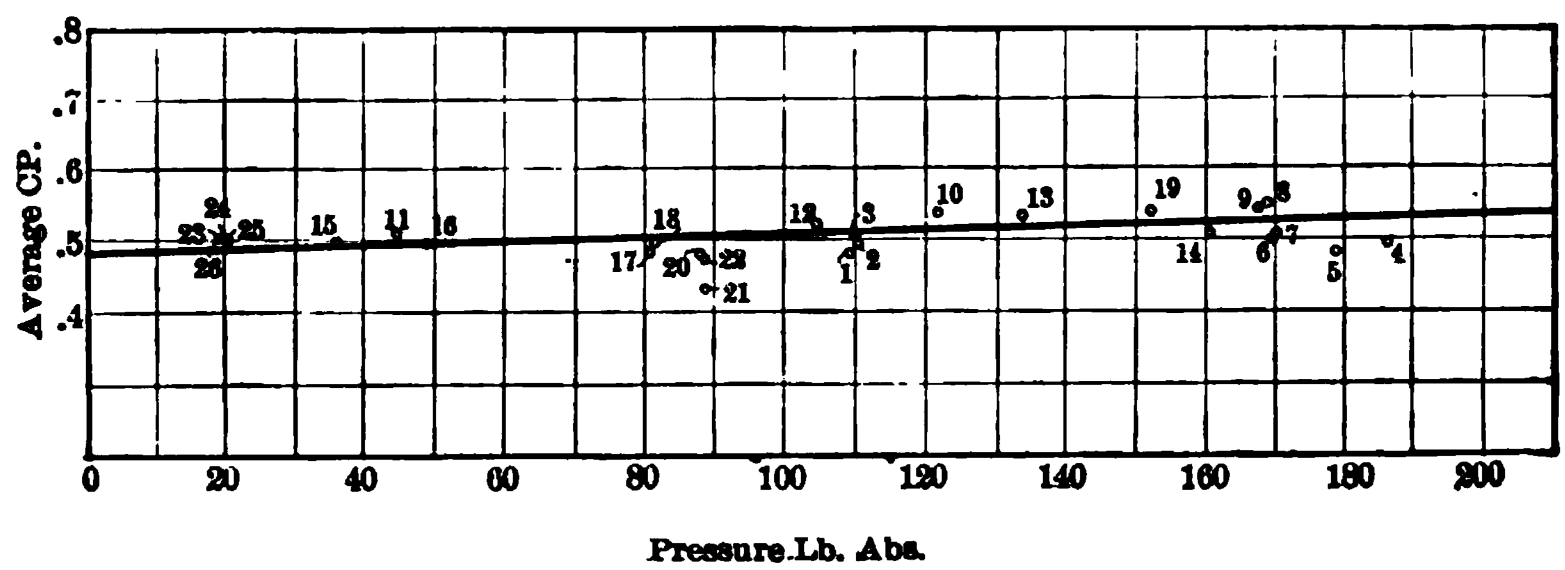


FIG. 83. Specific Heat of Superheated Steam, A. R. Dodge.

The maximum figure ranges as high as 0.8 and the minimum 0.48 for a given pressure and degree of superheat.

The curves in Fig. 82 are based upon the experiments of Knoblauch and Jakob ("Mitteilungen über Forschungsarbeiten," etc., Heft 36, p. 109, and *Stevens' Indicator*, October, 1905); those in Fig. 83 upon the

experiments of A. R. Dodge (Trans. A.S.M.E., 1907); those in Fig. 84 are plotted from tests of Burgoon, Carpenter, and Thomas (Trans. A.S.M.E, 1907); and those in Fig. 85 are based upon the investigation of Professor Thomas (Trans. A.S.M.E., December, 1907). These curves

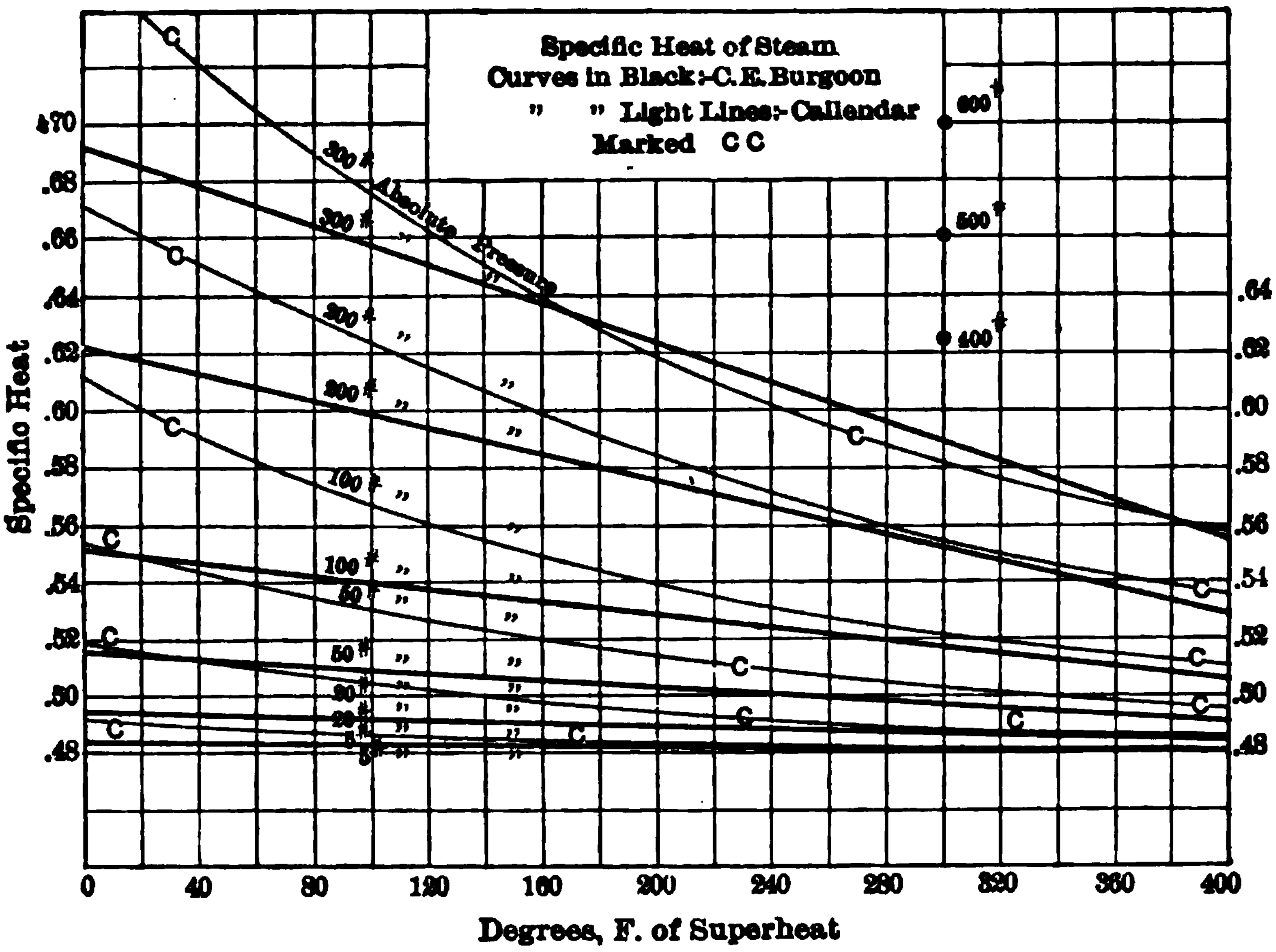


FIG. 84. Specific Heat of Superheated Steam, C. E. Burgoon.

differ both in theory and in value of c_p, but until further experiments prove otherwise the values in Fig. 82 may be accepted as sufficiently accurate for all practical purposes. The values given by Knoblauch and Jakob have been accepted by authorities as the most reliable. Table 17a is based upon their results. Table 17b has been calculated by means of Linde's equation. (See footnote, page 153.)

TABLE 17.

VALUE OF c_p AT ATMOSPHERIC PRESSURE BY VARIOUS AUTHORITIES.

Superheated Steam Cooled by Water-Jacketed Calorimeter.

Author.	Publication and Date.	Temp. Deg. F.	c_p at Atmos. Pres.	Variation of c_p with Increasing Pressure.	Variation of c_p with Increasing Temp.
Regnault	Ann. de Chimie et de Physique, Tome 23.	Varied	0.4805	None	None
Carpenter (Jones)	Sibley Journal, 5–1904.	Varied	0.4844	Increases	None
Dodge	Trans. A.S.M.E.	Varied	0.48	Increases	None

TABLE 17 — *Continued.*

Throttling Calorimeter. Saturated Steam Expanded to Lower Pressure.

Author.	Publication and Date.	Temp. Deg. F.	c_p at Atmos. Pres.	Variation of c_p with Increasing Pressure.	Variation of c_p with Increasing Temp.
Grindley	Phil. Trans., Vol. 194.	239	0.4317	Increases	Increases
Hirn				Increases	Decreases
Griessmann	Zeit. V. D. Ing., 52, 1903.	269	0.506	Increases	Increases
Peake	Proc. Royal Soc. A-509, 1905.	Varied	0.43	Increases	Increases
Carpenter (Stewart and Marble)	Sibley Journal, May, 1904.	Varied	0.463	Increases	None
Carpenter (Hoxie and Wood).	Sibley Journal, May, 1904.	Varied	0.4825	Increases	None
Carpenter (Sickles).	Sibley Journal, May, 1904.	Varied	0.48	Increases	None

Superheating Steam Electrically.

Author.	Publication and Date.	Temp. Deg. F.	c_p at Atmos. Pres.	Increasing Pressure.	Increasing Temp.
Peake	Proc. Royal Society A509, 1903.	Varied	0.46	None	None
Carpenter (Berry)		Varied	0.48	Increases	None
Carpenter (Thomas).	Trans. A.S.M.E., Eng. Mag., March 1907.	212 402	0.49 0.487	 Increases	 Decreases
Lorenz	Z.V.D.I., No. 20			Increases	Decreases
Knoblauch and Jakob.	Engineering, L, Feb. 22, 1907.	212 700	0.445 0.49	 Decreases	Decreases then increases

From Combustion of Explosive Gases.

Author.	Publication and Date.	Temp. Deg. F.	c_p at Atmos. Pres.	Increasing Pressure.	Increasing Temp.
Mallard and LeChatelier.	Zeit. V. D. Ing., Tome 48.	212	0.46	None	Increases
Sarran and Vieille	Do	212	0.464	None	Increases
Langen	Do	212	0.463	None	Increases

From Calculation.

Author.	Publication and Date.	Temp. Deg. F.	c_p at Atmos. Pres.	Increasing Pressure.	Increasing Temp.
Reeve	Wor. Poly. Journal	215	0.39	Increases	Increases
Hirn		265	0.4895	Increases	Decreases
Gray	London Engineer	236	0.38		
Zeuner			0.568		
Weyrauch	Zeit. V. D. Ing., Tome 48.	212	0.468	Decreases	Increases
Perry	Steam Engine	212	0.36	Increases	None
Roentgen	Thermodynamics	Varied	0.4805	None	None
Wagner	Rose Technic, 1905	284.4	0.513	Increases	
Knoblauch, Linde, and Klebe	Publication by authors, Berlin, 1905	212 356	0.493 0.479	Increases	Decreases

TABLE 17a.

MEAN SPECIFIC HEAT OF SUPERHEATED STEAM.

(Based upon the Experiments of Knoblauch and Jakob.)

Absolute Pressure, Pounds per Square Inch.	Degrees of Superheat, Fahr.																			
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
1	.452	.452	.453	.453	.454	.454	.454	.455	.455	.455	.455	.455	.455	.456	.456	.456	.456	.456	.456	.457
5	.460	.460	.460	.460	.460	.460	.459	.459	.459	.459	.459	.459	.459	.459	.459	.459	.460	.460	.460	.460
10	.465	.465	.465	.465	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.465	.465	.465	.465
15	.470	.470	.470	.470	.470	.470	.469	.469	.469	.469	.469	.469	.469	.469	.469	.468	.468	.468	.468	.468
20	.475	.475	.475	.475	.474	.474	.474	.474	.473	.473	.473	.473	.473	.472	.472	.472	.472	.472	.471	.471
25	.480	.480	.480	.479	.479	.479	.478	.478	.478	.477	.477	.477	.477	.477	.476	.476	.476	.475	.475	.475
30	.485	.485	.484	.484	.484	.483	.483	.483	.483	.482	.482	.481	.481	.481	.480	.480	.479	.479	.478	.477
35	.490	.490	.489	.489	.488	.488	.487	.487	.486	.486	.486	.485	.485	.484	.484	.483	.482	.482	.481	.480
40	.495	.494	.494	.493	.492	.492	.491	.491	.490	.490	.489	.489	.488	.488	.487	.486	.485	.484	.483	.482
50	.509	.508	.507	.506	.504	.503	.501	.500	.499	.498	.497	.496	.496	.495	.494	.493	.491	.490	.489	.487
75	.540	.535	.533	.530	.528	.525	.523	.521	.519	.517	.515	.513	.512	.511	.509	.506	.504	.502	.500	.497
100	.570	.560	.556	.552	.550	.546	.542	.540	.537	.534	.531	.528	.526	.524	.522	.518	.515	.512	.509	.505
125	.590	.585	.580	.575	.570	.565	.560	.556	.552	.548	.545	.542	.539	.537	.534	.532	.524	.520	.517	.510
150	.620	.615	.606	.597	.592	.585	.579	.572	.566	.562	.558	.554	.550	.547	.544	.539	.533	.528	.524	.518
175	.660	.645	.633	.622	.614	.605	.597	.589	.582	.577	.572	.566	.562	.558	.555	.548	.541	.535	.531	.524
200	.690	.675	.657	.645	.634	.623	.614	.605	.596	.590	.584	.578	.572	.568	.563	.556	.548	.542	.537	.529
225	.730	.710	.686	.672	.656	.643	.631	.620	.611	.603	.595	.589	.583	.578	.573	.562	.553	.547	.543	.535
250	.770	.740	.712	.695	.678	.663	.648	.635	.624	.615	.606	.599	.593	.587	.582	.570	.562	.555	.549	.540
275	.800	.770	.746	.722	.700	.681	.664	.650	.638	.629	.620	.610	.602	.596	.590	.578	.569	.561	.555	.545
300	.850	.805	.773	.740	.724	.702	.683	.666	.652	.641	.630	.621	.612	.605	.599	.586	.576	.568	.561	.550

TABLE 17b.

SPECIFIC VOLUME OF SUPERHEATED STEAM.

Cubic Feet per Pound.

(Knoblauch, Linde and Klebe.)

Absolute Pressure, Pounds per Square Inch.	Degrees of Superheat, Fahr.																			
	0	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250
1	333.0	339.0	345.0	352.0	358.0	364.0	370.0	376.0	331.0	388.0	394.0	400.0	406.0	412.0	418.0	424.0	439.0	454.0	468.0	483.0
5	73.3	74.6	75.9	77.2	78.4	79.7	80.9	82.1	83.3	84.5	85.7	86.9	88.2	89.4	90.6	91.8	94.8	97.8	101.0	104.0
10	38.4	39.0	39.7	40.3	40.9	41.5	42.2	42.8	43.4	44.0	44.6	45.2	45.8	46.4	47.0	47.7	49.2	50.7	52.2	53.7
15	26.3	26.7	27.1	27.6	28.0	28.4	28.8	29.2	29.6	30.0	30.5	30.9	31.3	31.7	32.1	32.5	33.5	34.5	35.5	36.6
20	20.1	20.4	20.7	21.0	21.4	21.7	22.0	22.3	22.6	22.9	23.3	23.6	23.9	24.2	24.5	24.8	25.5	26.3	27.0	27.8
25	16.3	16.6	16.8	17.1	17.3	17.6	17.8	18.1	18.4	18.6	18.8	19.1	19.4	19.6	19.8	20.1	20.7	21.3	21.9	22.6
30	13.7	14.0	14.2	14.4	14.6	14.8	15.0	15.2	15.5	15.7	15.9	16.1	16.3	16.5	16.7	16.9	17.4	18.0	18.5	19.0
35	11.9	12.1	12.3	12.5	12.7	12.9	13.0	13.2	13.4	13.6	13.7	13.9	14.1	14.3	14.5	14.6	15.0	15.5	15.9	16.4
40	10.5	11.2	11.4	11.5	11.7	11.9	12.1	12.2	12.4	12.6	12.7	12.9	13.1	13.2	13.4	13.6	14.0	14.4	14.8	15.2
50	8.5	8.7	8.8	8.9	9.0	9.2	9.3	9.4	9.6	9.7	9.8	9.9	10.1	10.2	10.3	10.5	10.8	11.1	11.4	11.7
75	5.81	5.91	6.00	6.10	6.19	6.28	6.37	6.46	6.55	6.64	6.73	6.82	6.91	7.00	7.09	7.17	7.38	7.60	7.81	8.02
100	4.43	4.51	4.58	4.65	4.72	4.79	4.86	4.93	5.00	5.07	5.14	5.21	5.27	5.34	5.41	5.47	5.63	5.80	5.96	6.12
125	3.58	3.64	3.70	3.76	3.82	3.88	3.94	4.00	4.06	4.11	4.17	4.22	4.28	4.33	4.39	4.45	4.58	4.71	4.84	4.97
150	3.01	3.06	3.11	3.17	3.22	3.27	3.32	3.37	3.42	3.46	3.51	3.56	3.61	3.66	3.70	3.75	3.86	3.97	4.08	4.19
175	2.60	2.65	2.69	2.74	2.78	2.83	2.87	2.91	2.96	3.00	3.04	3.08	3.12	3.16	3.20	3.24	3.34	3.44	3.53	3.63
200	2.29	2.33	2.37	2.41	2.45	2.49	2.53	2.57	2.61	2.64	2.68	2.72	2.76	2.79	2.83	2.86	2.95	3.04	3.12	3.21
225	2.05	2.09	2.12	2.15	2.19	2.23	2.26	2.30	2.33	2.37	2.40	2.43	2.47	2.50	2.53	2.57	2.64	2.72	2.80	2.88
250	1.85	1.88	1.91	1.95	1.98	2.02	2.05	2.08	2.11	2.14	2.19	2.21	2.24	2.27	2.30	2.33	2.40	2.47	2.54	2.61
275	1.69	1.72	1.75	1.78	1.81	1.84	1.87	1.90	1.93	1.96	1.99	2.02	2.04	2.07	2.10	2.13	2.19	2.26	2.32	2.39
300	1.55	1.58	1.60	1.63	1.66	1.69	1.72	1.75	1.78	1.80	1.83	1.86	1.88	1.91	1.94	1.96	2.02	2.09	2.15	2.21

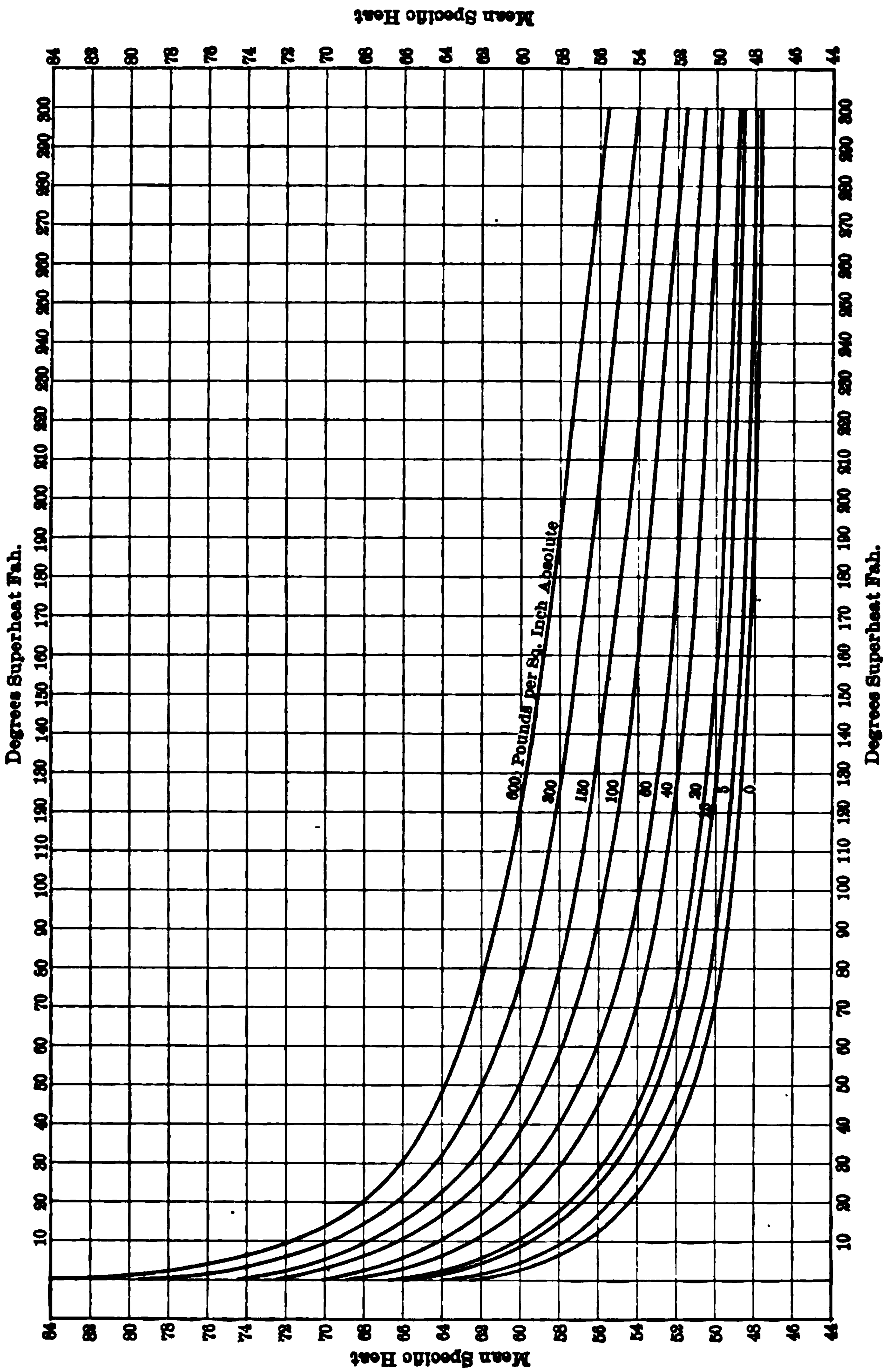


FIG. 85. Specific Heat of Superheated Steam, Carl C. Thomas.

Just how much these different values of the specific heat affect the calculated performance of an engine may be illustrated by the following case:

An engine uses 8.5 pounds of steam per I.H.P. hour, initial pressure 120 pounds absolute; back pressure 0.94 pound absolute; superheat at throttle 300 degrees F. Assume that the "heat of the liquid" in the exhaust steam is returned to the boiler and that the specific heat is 0.48 in one case and 0.8 in the other, these being the extreme values given by different experimenters for the given conditions.

	Case I.		Case II.	
Specific heat		0.48		0.8
B.T.U. in 1 pound of saturated steam above ideal hot-well temperature	1118.0		1118.0	
B.T.U. in superheat	144		240	
B.T.U. in 1 pound of superheated steam	1262.0		1358.0	
B.T.U. per I.H.P. per minute		162.1		192.4

On the basis of the specific heat of 0.48 the heat consumption, 162.1 B.T.U. per I.H.P. per minute, is exceptional and corresponds to an equivalent saturated steam consumption of 9.6 pounds per I.H.P. hour, whereas the performance of 192.4 B.T.U. per I.H.P. per minute based upon the specific heat of 0.8 has been excelled by a number of actual engines using saturated steam.

106. Superheaters. — The installation of a superheater is equivalent to an increase in boiler capacity. The superheater may be independently fired or can be arranged in connection with the boiler proper. The heating surface is usually of wrought iron, mild steel, cast iron, or cast steel. Engineers are not agreed as to which arrangement or which material gives the most economical returns. The recent symposium on superheated steam conducted by the research committee of the American Society of Mechanical Engineers (Trans. A.S.M.E., 1907) clearly indicated this lack of agreement. The requirements for a successful superheater are:

1. Security in operation, or minimum danger of overheating.
2. Economical use of heat applied.
3. No exposure of joints to the fire.
4. Provision for free expansion.
5. Disposition such that it may be cut out or repaired without interfering with the operation of the plant.
6. Ease of application to existing plants.

Nearly all superheaters depend upon carrying steam at a high velocity through small tubes in the form of return bends or coils and arranged to be heated by the hot gases in the boiler furnace or from some other source.

The independently fired superheater has the following advantages:

1. The degree of superheat may be varied independently of the performance of the boiler.
2. It can be placed at any desirable point.
3. Repairs are readily made without shutting down boilers.

Some of the disadvantages are:

1. It requires separate firing and extra attention.
2. Saturated steam can only be furnished in case of a breakdown to the superheater.
3. Extra piping is required.
4. Extra space required.

Standard practice in this country advocates that the superheater be contained within the boiler setting. Of two hundred recent installations, one hundred and eighty, or ninety per cent, were of this type.

Fig. 86. Babcock and Wilcox Superheater.

107. Babcock & Wilcox Superheater. — Fig. 86 shows the application of superheating coils to a Babcock & Wilcox boiler, illustrating the *indirectly fired* type. The wrought-iron tubes are bent into U shape, the ends being connected into manifolds, the upper one receiving the saturated steam from the boiler and the lower one the superheated

steam after it has traversed the superheater tubes. A small pipe connects the lower manifold with the water space of the boiler by means of which the superheater may be cut out if desired, or flooded when starting up. Any steam formed in the superheater tubes is returned into the boiler drum through the collecting pipe, which, when the superheater is at work, conveys saturated steam into the upper manifold. When steam pressure has been attained the superheater is thrown into action by draining the water away from the manifolds and opening the superheater stop valve. The tubes are free at one end and the manifolds are not rigidly connected with each other, thus avoiding expansion strains. With the proportion of superheating surface to boiler surface ordinarily adopted the steam is superheated from 100 to 150 degrees F.

FIG. 87. Stirling Superheater.

108. Stirling Superheater. — This superheater consists of two drums, Fig. 88, connected by seamless drawn tubes two inches in diameter. It may take the place of the middle bank of tubes in the Stirling boiler as shown in Fig. 87, or be installed in the final pass of the gases in the back

of the boiler. The drums around the tubes are protected from intense heat by asbestos cement. A pipe connecting the front drum of the boiler with the lower drum of the superheater permits the coils to be flooded in starting up or when the superheater is not needed. In this

FIG. 88. Arrangement of Tubes; Stirling Superheater.

case the superheater acts as additional boiler-heating surface. The upper drum is divided into three and the lower into two compartments. The tubes are arranged with alternately wide and narrow spacing, so that any tube may be removed without disturbing the rest. The flow of steam is indicated by arrows.

109. Foster Superheater. — Fig. 89 shows the application of a Foster superheater to a Babcock & Wilcox boiler. This device consists of cast-iron headers joined by a bank of straight parallel seamless drawn-steel tubes, each tube being encased in a series of annular flanges placed close to each other and forming an external cast-iron covering of large surface. The tubes are double, the inner tube serving to form a thin annular space through which the steam passes as indicated. Caps are provided at the end of each element for inspection and cleaning purposes. Foster superheaters are more costly than plain-tube superheaters, but are longer lived and offer a much larger heating surface in proportion to the space occupied.

Fig. 91 shows a Foster superheater arranged for independent firing.

The "Schwoerer" superheater, which is somewhat similar in external appearance to the Foster, differs from it considerably in detail, the heating surface being made up of suitable lengths of cast-iron pipe ribbed outside circumferentially and inside longitudinally. The ends of the pipes are flanged and connected by cast-iron U-bends. The intention is to provide ample heating surface internally and externally, with a compact apparatus.

Fig. 89. Foster Superheater in Babcock & Wilcox Boiler.

110. Independently Fired Superheaters. — The Schmidt superheater, Fig. 90, consists of two nests of coils, *A* and *D*, of equal size and dimensions, connected to cast-iron headers *O* and *I*. Saturated steam enters the first nest of coils through *C* and passes into header *O*. From *O* the steam, which is now dried and partly superheated, flows through the cast-iron pipe *E* to header *I*, and thence through the second nest of coils into header adjoining *O*, and through pipe *R* to the engine. In chamber *D* the steam and gases flow on the counter-current and in chamber *A* on the concurrent principle. This combination permits of a low flue temperature and high steam temperature without subjecting the tubes to an excess of heat as would be the case if the steam left the coils *A* at header *I*, where the furnace gases are the hottest. A steam

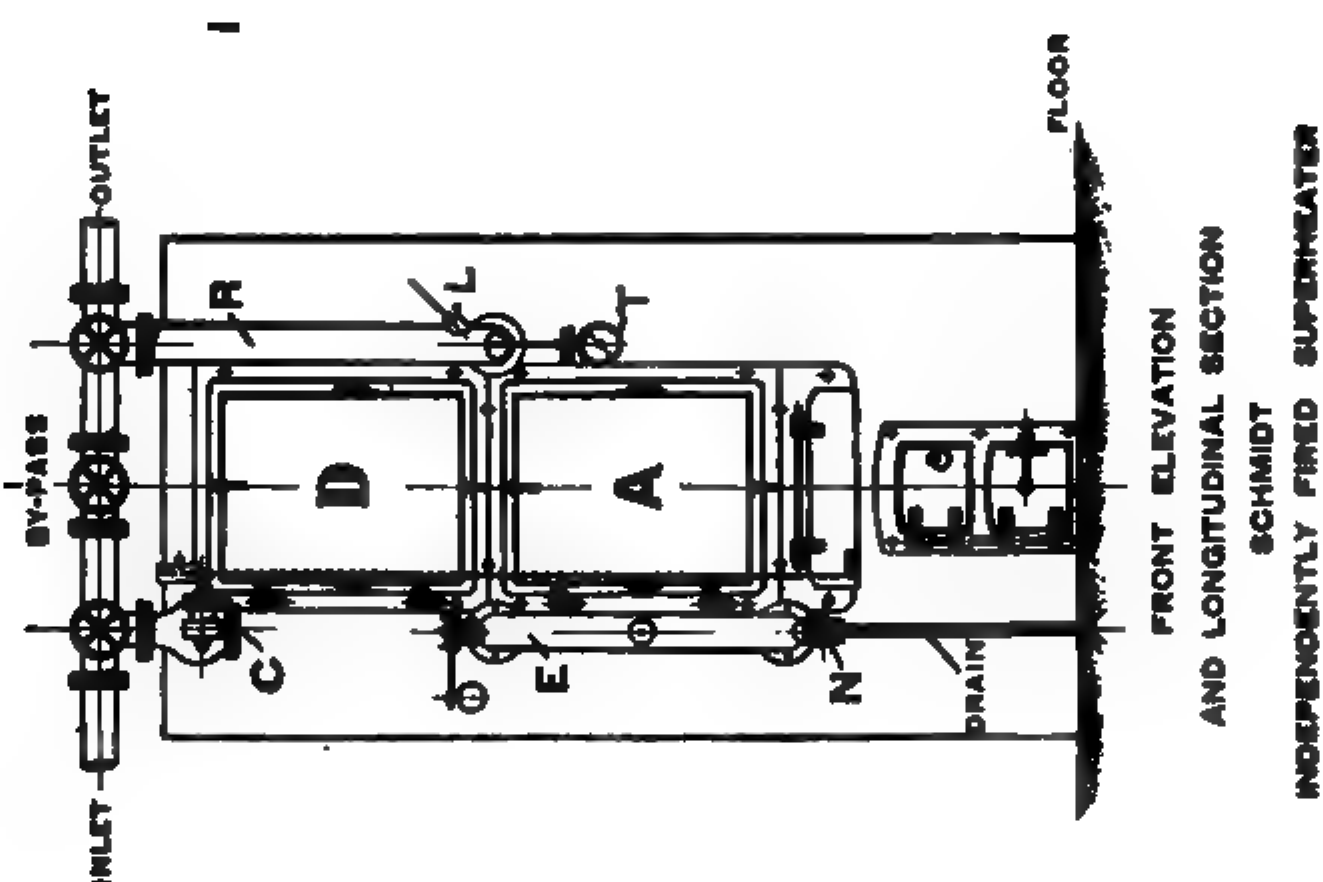


Fig 90. Schmidt Independently Fired Superheater.

INLET

Fig. 91. Foster Independently Fired Superheater.

temperature of 750 degrees F. and a flue temperature of 450 degrees F. are easily maintained with this apparatus. A mercury pyrometer T is fitted where the superheated steam enters the discharge pipe R. A thermometer cup L permits of checking the pyrometer by means of a nitrogen-filled thermometer. Each coil can be taken out separately and a new one put in without removing the others or dismantling the plant. Water produced by condensation while the superheater is inoperative collects in the bottom header N and escapes through a drain cock. If the steam supply should be suddenly shut off, the air door P is opened automatically by weight K. As soon as steam begins to flow it raises the weight through the opening of valve C and the door closes. The Schmidt superheater when arranged in the flue has practically the same construction as the independently fired.

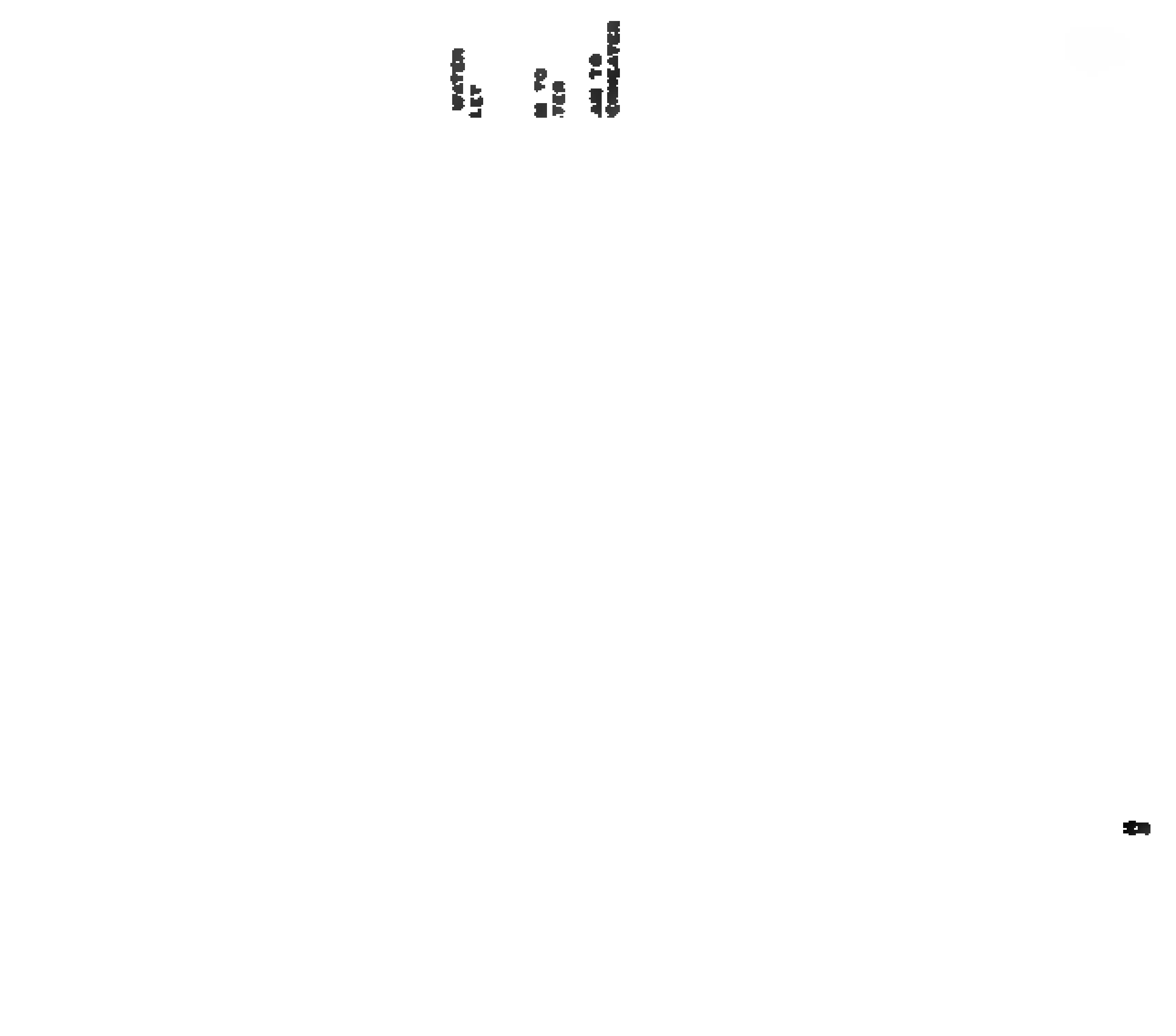


FIG. 92. Schmidt System of Combined Superheater, Feed Water Heater and Economizer.

Fig. 92 shows a combination of Schmidt superheater, economizer, and feed-water heater which finds much favor with engineers on the Continent.

111. Materials used in Construction of Superheaters. — Most superheaters are constructed either of wrought iron, mild steel, cast iron, or cast steel, the latter material having the advantage of not being damaged by any temperature to which it is likely to be subjected, which does away with the necessity of damper mechanisms and simplifies the installation. On the other hand, cast-metal superheaters are usually ribbed after the fashion of an air-cooled gas engine, and are, therefore, very heavy and thick walled, necessitating a higher temperature for the same useful effect than in the case of the wrought-iron construction, but have the advantage of minimizing fluctuation of steam temperature which would otherwise be caused by a wide variation in temperature of furnace. One of the most successful cast-metal heaters is of European design and is constructed of a special alloy known as "Schwoerer" iron. Table 18 gives the yearly cost of repairs to piping and necessary brickwork for a number of installations equipped with cast-metal superheaters of the "Schwoerer" type.

Wrought iron and mild steel offer the advantage of lightness, ease of construction, and low first cost, but cannot be exposed to very high temperatures without injury, and consequently provision must be made for diverting the direction of the heated gases or for flooding the coils while the boiler is being warmed before steam is generated.

Neither cast iron nor steel loses in tensile strength when subjected for a very short time to the temperature of superheated steam, but, on the contrary, may be stronger. Tests made by Professor Lanza ("Applied Mechanics," p. 489) showed that the tensile strength of steel diminished from 0 degrees F. to about 300 degrees F. and then increased, reaching a maximum between 500 and 650 degrees F. Cast iron and steel maintained their strength, with a tendency to increase, up to 900 degrees F. beyond which the strength is diminished.

Ordinary cast-iron valves and fittings have shown permanent increase in dimensions under high superheat and in numerous instances have failed altogether, but sufficient data are not available to prove conclusively the unreliability of cast iron if the iron mixture is properly compounded and the necessary provision is made for expansion and contraction.*

112. Extent of Superheating Surface. — The required extent of superheating surface for any proposed installation depends upon (1) the degree of superheat to be maintained; (2) the velocity of the steam through the superheater; (3) the character of the superheater; (4) the weight of the steam to be superheated; (5) the moisture in the wet steam; (6) the temperature of the gases entering and leaving the superheater; and (7) the conductivity of the material.

* See Symposium on the "Effect of Superheated Steam on Cast Iron and Steel," Jour. Am. Soc. Mech. Engrs., Dec., 1909.

TABLE 18.

AVERAGE YEARLY EXPENSES FOR REPAIRS FOR CAST-IRON SUPERHEATERS ("SCHWOERER" TYPE) AS OBTAINED FROM SIX PLACES OF INSTALLATION.

(Otto Berner, *Power*, August, 1904.)

Number.	Length of Time of Installation. Years.	Average Daily Use. Hours.	Place of Installation of Superheater with Reference to Boiler.	Average Temp. of Gases immediately in Front of the Superheater Surface. Degrees F.	Average Steam Pressure. Pounds per Square Inch (Absolute).	Average Temp. of Steam. Degrees F.	No. of Elements for One Superheater.	Length of One Element in Feet.	Average Yearly Cost of Repairs. 1 Superheater.	Average Yearly Cost of Repairs. For 1 Foot of Element.
1	8	11	Directly heated........	....	99.56	518°	12	10.5	$71.40	$0.61
2	5	12	Directly behind the fire bridge............	....	106.672	....	4	9.843	$14.09	$0.37
3	10	11	a. Directly behind the fire bridge........ b. Behind the first flue.	.... 716°–842°	78.232 113.79	.... 442.4°	2 6	8.202 9.843	11.90 0	$0.68 0
4	11	12	Behind the first flue.....	....	85.34	392°–572°	6	9.843	$8.33	$0.15
5	6	24	Behind the first flue (fire-tube boiler)........	....	99.56 & 156.46	482°	8 to 10	6.562	$5.95	$0.08
6	7	11	Directly behind the fire bridge............	....	170.68	500°–572°	....	....	0	0
Average Results	7.7	13.5	....	....	113.79	493°	6.9	5.25	$15.95	$0.28

Since the heat absorbed by the steam in the superheater is equal to that given up by the products of combustion, neglecting radiation, this relationship may be expressed

$$SUd = Wc\,(t_1 - t_2), \tag{23}$$

in which

S = square feet of superheating surface per boiler horse power.

U = coefficient of heat transmission, B.T.U. per square foot per hour per degree difference in temperature.

d = mean temperature difference between the steam and heated gases, degrees F.

W = weight of gases passing through the superheater per boiler horse power hour.

c = specific heat of the gases.

t_1 = temperature of the gases entering superheater, degrees F.

t_2 = temperature of the gases leaving superheater, degrees F.

Transposing equation (23),

$$S = \frac{Wc\,(t_1 - t_2)}{Ud}, \tag{24}$$

On account of the great variation in the values given for U, and the difficulty of determining d, t_1, and t_2 for different types of superheaters, equation (24) is hardly applicable in practice.

An empirical formula for determining the extent of superheating surface in connection with indirect superheaters which appears to conform with practice is given by J. E. Bell (Trans. A.S.M.E., May, 1907):

$$x = \frac{10\,S}{2\,(T - t) - S}, \tag{25}$$

in which

x = square feet superheating surface per boiler horse power.

S = superheat, degrees F.

T = temperature of the products of combustion where the superheater is located.

t = temperature of the saturated steam.

The value of T may be found from the equation

$$\frac{1}{(T - t)^{0.16}} = 0.172\,H + 0.294, \tag{26}$$

in which

H = the per cent of boiler-heating surface between the point at which the temperature is T and the furnace.

t as in (25).

Fig. 92a. Curve showing Relation between Gas Temperature, and Extent of Heating Surface passed over.

Equation (26) is based upon the assumption that the heat transferred from the gases to the water is directly proportional to the difference in temperature; that the furnace temperature is 2,500 degrees F.; flue temperature 500 degrees F.; steam pressure 175 pounds per square inch gauge; one boiler horse power is equivalent to 10 square feet of water-heating surface.

Example: What extent of heating surface is necessary to superheat saturated steam at 175 pounds gauge pressure, 200 degrees F., if the superheater is placed in the boiler setting where the gases have already traversed 40 per cent of the water-heating surface?

Substitute $H = 0.4$ and $t = 378$ in equation (26),

$$\frac{1}{(T - 378)^{0.16}} = 0.172 \times 0.4 + 0.294$$

$$T = 950.$$

Substitute $T = 950$ and $S = 200$ in equation (25),

$$x = \frac{10 \times 200}{2\,(950 - 378) - 200}$$

$$= 2.12 \text{ square feet.}$$

The curve in Fig. 92a was plotted from equation (26), and gives a ready means of determining T and of observing the law governing heat absorption by the boiler between furnace and breeching. The abscissas represent the temperatures of the hot gases at different points in their path between furnace and breeching. The ordinates represent (1) the per cent of boiler-heating surface passed over by the hot gases, and (2) the per cent of the total heat generated which is absorbed by this heating surface.

In the use of equation (26) the probability of error is greatest when considering a point near the furnace, since large quantities of heat are transmitted to the tubes by radiation from the fuel bed which are not taken account of. For most practicable purposes the assumption is sufficiently accurate.

For the application of the curve in Fig. 92a to the design of direct and indirect superheaters for various degrees of superheat, see "Stirling," published by the Stirling Boiler Company, pp. 92–96.

113. Performance of Superheaters. — Published tests of both directly and indirectly fired superheaters cover such a wide range of conditions of installation and operation that general conclusions cannot be drawn, but it may be of interest to note briefly the performances in a few specific cases.

The curves in Figs. 93, 94, and 95 are plotted from tests of a Babcock & Wilcox boiler, with 5000 square feet of water-heating surface, equipped with superheating coils of 1000 square feet area, as illustrated in Fig. 62. The furnace with ordinary short ignition arch was provided with chain grate of 75 square feet area.

Fig. 93 shows the relation between degrees of superheating and total horse power of boiler and superheater.

Fig. 94 shows the relation between the horse power produced in the boiler and the percentage of boiler horse power produced in the superheater.

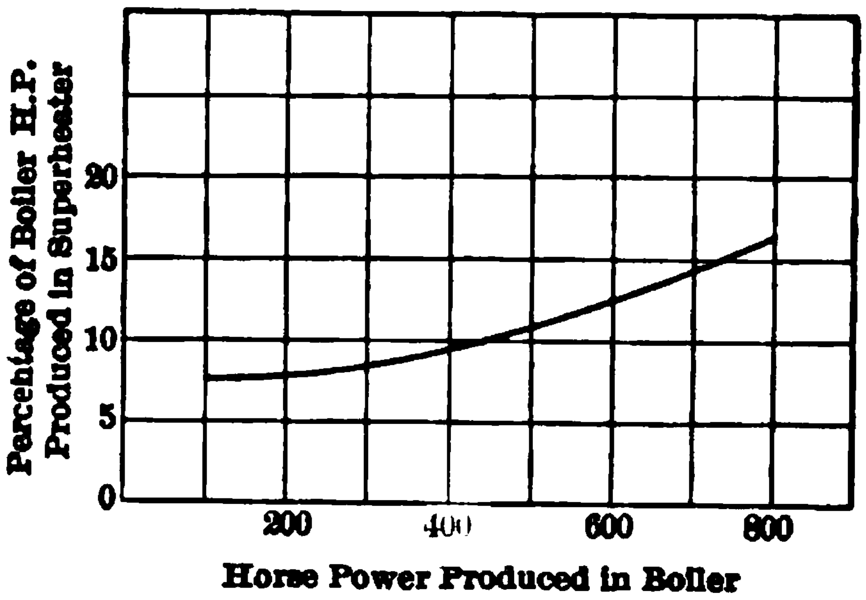


Fig. 94. Percentage of Horse-Power Produced in the Superheater of that Developed in the Boiler.

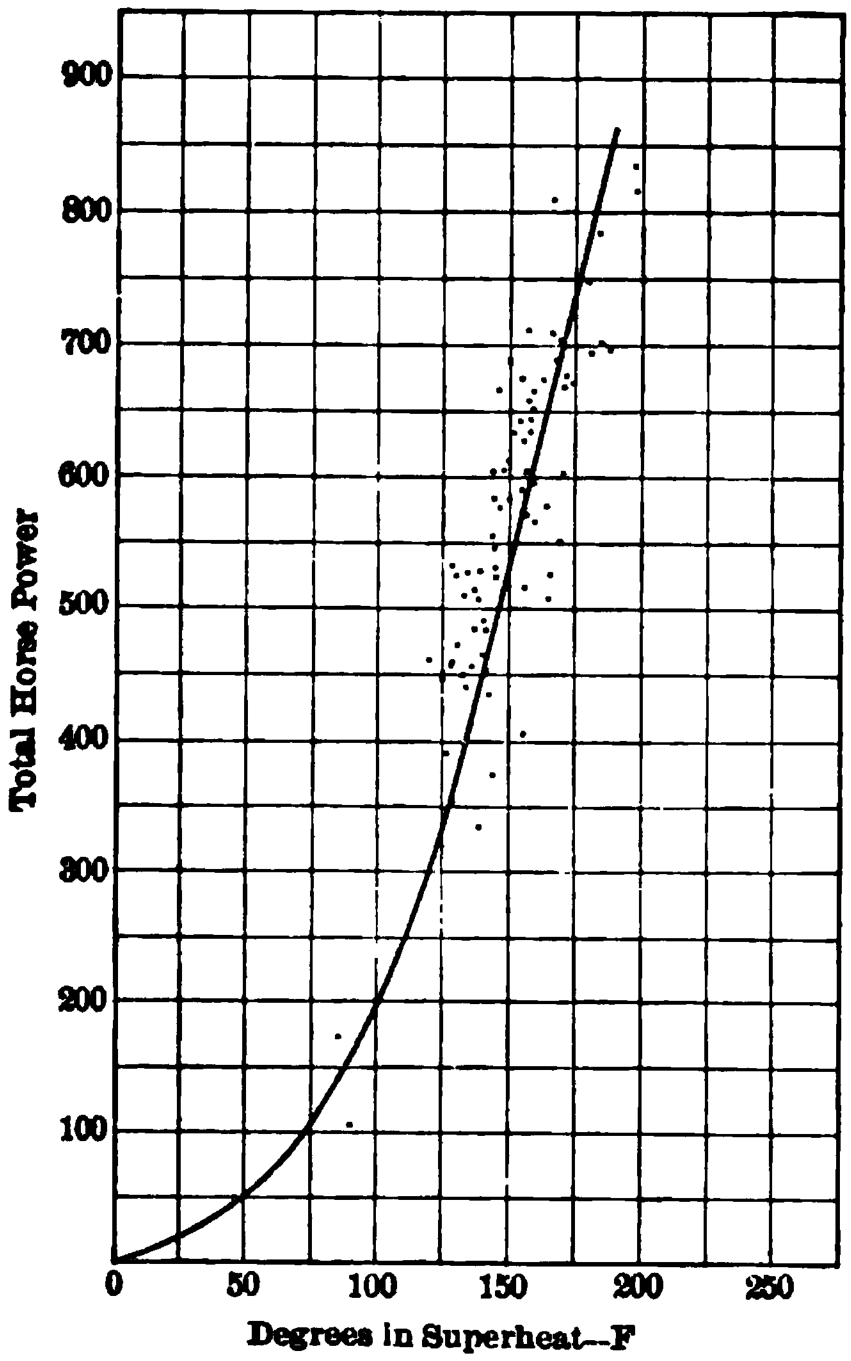


Fig. 93. Relation of Degree of Superheat to Total Horse-Power Developed.

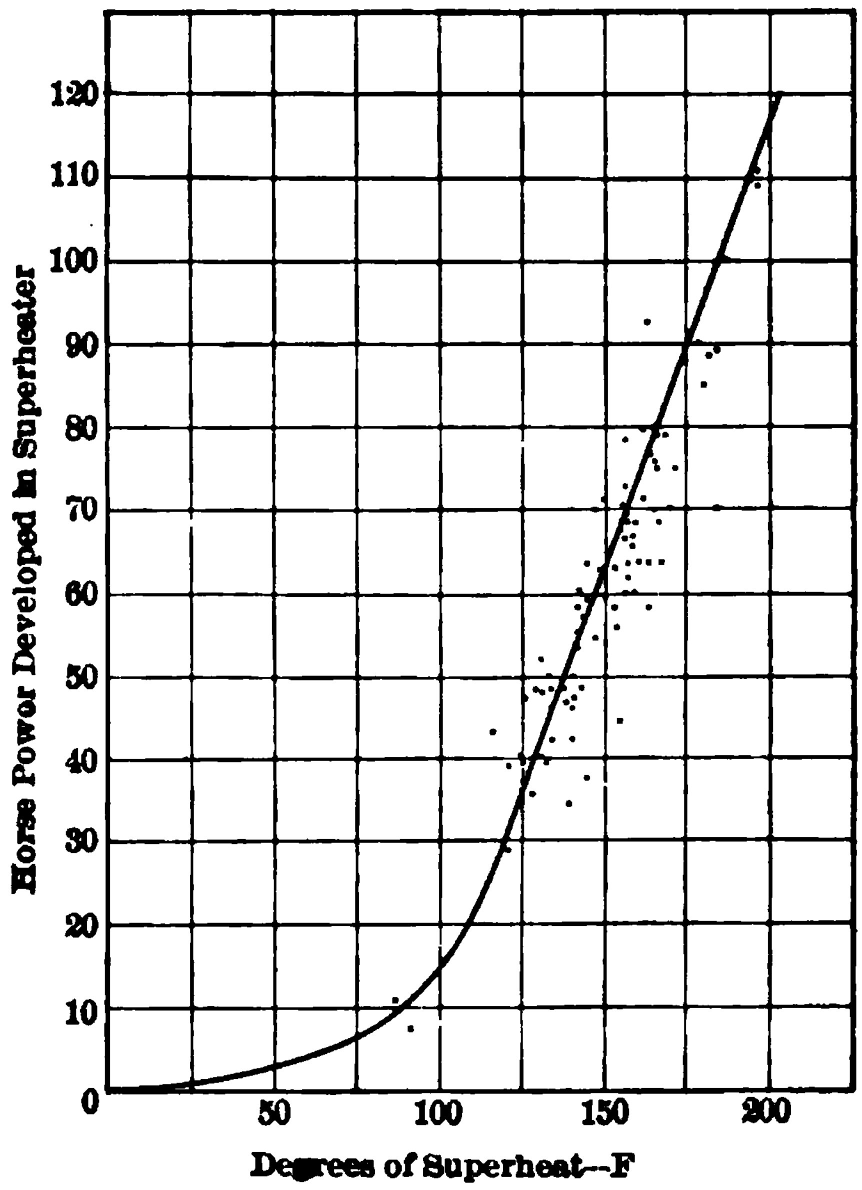


Fig. 95. Relation of Degree of Superheat to Horse-Power of Superheater.

Fig. 95 shows the relation between the degree of superheat obtained and the horse power developed in the superheater.

TABLE 19.

DIFFERENCE IN HEAT EFFICIENCY OF SUPERHEATERS INSTALLED IN FLUE AND SEPARATELY FIRED SUPERHEATER.
(Otto Berner, *Power*, August, 1904.)

Style of Superheater	Separately Heated Superheater. (System Uhler.)		Boiler-Flue Superheater. (Patent Schwoerer)	
Number of test	1	2	3	4
Heating surface of superheater . . . Square feet	1,636.17	1,636.17	2,744 90	1,829.93
Number of boilers used in test	3	2	3	2
Boiler-heating surface . . . Square feet	4,843.94	3,229 29	4,843.94	3,229 29
Grate area . . . Square feet	104.28	69.52	104.28	69.52
Feed water per square foot of boiler-heating surface. Pounds per hour	1.96	2.75	2.04	2.8
Coal consumed per square foot of grate area . . . Pounds per hour	9.31	14.05	9.96	14.48
Boiler pressure, absolute . . . Pounds per square inch	177.48	175.64	179 91	176.35
Feed-water temperature . . . Degrees F.	64 76	62.6	50.00	50.00
Steam temperature on leaving superheater . . . Degrees F.	483.8	508.46	512.6	550.4
Hea[illegible] per [illegible] { Boiler . . . B.T.U.	11,553 39	11,187 46	. . . . . .	. . . . . .
[illegible] . . . B.T.U.	4,876 81	5,544.76	. .	. . . . . .
[illegible] . . . B.T.U.	10,880 53	10,605 30	12,392.33	12,229 88
Hea[illegible] le[illegible] d superheater . . . Per cent	5.70	5.33	. . . . .	. . . . .
Hea[illegible] r installed in flues greater than when using boiler only . . . Per cent	. . . . .	. . . .	6.87	8.75
Heat efficiency when using flue superheater greater than when using separately fired superheater . . . Per cent			13.88	15.72

TABLE 90.

DECREASE IN TEMPERATURE OF GASES OF COMBUSTION DUE TO SUPERHEATER INSTALLED IN FLUE.

(Otto Berner, *Power*, August, 1904).

Test	1		2		3		4		5		6	
Style of boiler	Water-Tube Boilers.											
Boiler-heating surface. Square feet	3,584.51		2,884.83		2,529.61		1,955.86		3,089.35		936.49	
Superheater surface. Square feet	....		344.45		645.85		Schwoerer Patent		1,054.9		182.99	
Grate area. Square feet	81 8		51.02		55.97		45 21		64.58		26.47	
Running of boiler	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.
Feed-water consumption per square foot of heating surface. Pounds per hour	2.46	2.23	2.68	2.66	2.91	2.69	2.31	2.10	2.47	2.03	3.70	3.50
Coal consumption per square foot of grate area. Pounds per hour	14.23	13.55	19.57	19.74	27.84	26.00	15.7	14.02	14.54	11.79	26.32	18.51
Boiler pressure (absolute)	130.8	131.51	149.6	148.38	182.68	181.75	122.99	121.73	149.33	150.75	149.33	151.31
Feed-water temperature. Degrees F.	107.6	104.72	43.97	45.68	89.9	89.6	53.24	55.4	101.3	109.04	32	67.1
nbustion. Degrees F.	554	462.2	500	465.8	654.98	597.2	492.8	442.4	590	471.2	798.4	550.4
erheater. Degrees F	......	424.94	......	486.5	...	562.6	.....	489.7	......	536	....	486.5
Per cent	......	......	......	......	55.9	59.1	55.63	60.18	71.9	78.6	49.3	62.5
f plant due to .... Per cent	......	2.3	.....	3.3	....	5.4	.....	7.6	....	8.5	.....	21.2
irements due to use of superheated steam. Per cent	...	4.8	......	0.8	....	6.6	.	10.7	.	18.9	.....	23.7
Decrease in heating surface requirements due to use of superheated steam. Per cent	.....	9.4	...	0.9	....	9.3	.....	8.8	..	17.4	....	5.5

TABLE 21.

INCREASE IN HEAT EFFICIENCY OF THE BOILER DUE TO SUPERHEATER.

(Otto Berner, *Power*, August, 1904.)

	Boiler without Superheater.	Boiler with Superheater.	Boiler without Superheater.	Boiler with Superheater.	Boiler without Superheater.	Boiler with Superheater.	Boiler with out Superheater.	Boiler with Superheater.	Boiler without Superheater.	Boiler with Superheater.
Test	1	2	3	4	5	6	7	8	9	10
Boiler pressure (absolute).... Lb. per sq. in.	77.21	77.21	71.25	71.25	72.53	72.53	72.24	72.24	86.18	86.18
Feed-water temperature, Deg. F.	145.4	150.8	145.4	150.8	161.6	154.4	149.0	142.6	161.6	156.2
Steam temperature.....Deg. F.	....	361.4	....	363.2	....	372.2	....	366.8	....	372.2
Evaporation per square foot of heating surface. Lb. per hour	8.56	9.12	7.3	8.1	9.01	9.95	7.23	8.56	7.7	9.09
Coal consumption per square foot of grate area.......Lb. per hour	31.78	30.46	29.08	29.28	35.07	35.87	30.1	30.1	30.76	30.76
Increase in heat efficiency due to use of superheater..........Per cent	....	12.4	....	11.8	....	11.6	....	18.1	....	18.1

Tables 19 to 21 are taken from the report of Otto Berner ("Zeit. d. Ver. Deut. Eng." and reprinted in *Power* August, 1904).

Table 19 compares the heat efficiency of a steam plant equipped with directly and with separately fired superheaters, the former showing a much higher efficiency.

Table 20 compares different boilers with and without flue superheaters, showing the effect upon the temperature of the flue gases. The gain in heat efficiency of the entire plant due to the use of the superheater is decisive in each case.

TABLE 22.

(*Engineer, U. S.*, May 1, 1904.)

	With Superheater.	Without Superheater.
Time of start	12 noon, Feb. 8	11 A.M., Feb. 11
Time of finish	12 noon, Feb. 9	11 A.M., Feb. 12
Hours run	24	24
Average steam pressure	79.3 lb.	79.4 lb.
Average water pressure, triple expansion, head in feet	0.99	1.05
Average water pressure, compound, head in feet	7.10	7.10
Average vacuum of suction for triple and compound, inches of mercury	22.90	23.21
Total head on triple, feet of water	29.05	29.46
Total head on compound, feet of water	33.04	33.39
Total double strokes, triple	30,557	34,114
Total double strokes, compound	35,395	32,158
Gallons pumped from piston displacement, total, triple	2,854,023	3,186,247
Gallons pumped from piston displacement, total, compound	2,930,706	2,662,682
Gallons pumped from piston displacement, total, triple combined	5,784,720	5,848,930
Gallons, total, pumped as measured by weir	4,492,680	4,549,480
Per cent slip	22.3	22.2
Foot pounds, weir	1,163,815,819	1,184,983,596
Total coal consumed	5,015 lb.	6,410 lb.
Per cent refuse	23.7	18.7
Total refuse	1,188	1,203
Total feed water	38,399	50,960
Duty per 100 pounds coal	23,206,696	18,486,483
Duty per 1,000 pounds steam	30,308,498	23,253,213

Per cent increase of work per 100 pounds coal....................25.5
Per cent increase of work per 1,000 pounds steam................30.2
Per cent saving in coal per foot pound work......................20.2
Per cent saving in feed water per foot pound work................23.2
Average temperature steam leaving superheater..................527.4 deg. F.
Average temperature steam entering superheater.................320.1 deg. F.
Average degree superheat..207.3 deg. F.

Table 21 shows the gain in heat efficiency due to the use of superheaters in a number of plants equipped with fire-tube boilers.

Table 22 gives the results of tests on one of the return tubular boilers at the Spring Creek Pumping Station of the Brooklyn Waterworks (Feb. 9, 1904) with and without a superheater. The superheater, of the Foster type, was installed between the rear wall of the setting and the tube sheet.

113a. Properties of Superheated Steam. — The following equations derived by Prof. Goodenough of the University of Illinois and based upon the experiments of Knoblauch and Jakob, give a comparatively simple method of determining the various properties of superheated steam if steam and entropy tables are not available. The results as obtained from these equations agree substantially with Marks' and Davis' Steam Tables and the 1909 Edition of Peabody's Steam Tables.

T = absolute temperature of the superheated steam, deg. F.

p = absolute steam pressure, lbs. per sq. in.

λ = total heat, B.T.U. per pound.

u = intrinsic energy, ft.-lbs.

n = entropy.

C_p = *true* specific heat.

$$\lambda = T(0.372 + 0.00005\,T) - p\,(1 + 0.00035)\frac{C}{T^{3.5}} + 882.4,$$

in which $\log C = 9.42383$.

$$u = T(202.44 + 0.0389\,T) - \frac{C}{T^{3.5}}(1 + 0.00025\,p) + 686242,$$

in which $\log C = 12.20551$.

$$n = 0.85657 \log T + 0.0001\,T - 0.25392 \log p$$
$$- p\,(1 + 0.00035\,p)\frac{C}{T^{4.5}} - 0.4300,$$

in which $\log C = 9.31469$.

$$C_p = 0.372 + 0.0001\,T - p\,(1 + 0.00035\,p)\frac{C}{T^{4.5}},$$

in which $\log C = 9.96790$.

The *mean* specific heat may be obtained by subtracting the total heat of the saturated steam from that of the superheated steam and dividing the difference by the *degree* of superheat.

The specific volume may be determined from Linde's equation as stated at the bottom of page 153.

CHAPTER VI.

COAL AND ASH-HANDLING APPARATUS.

114. General. — The cost of coal and its delivery into the furnace are usually the largest items in the operating charges, hence large central stations are located, when practicable, adjacent to a railway line or water front, to minimize the cost of handling coal and ashes. Isolated stations in the business districts of large cities are usually unfavorably situated, so that the cost of handling coal and ashes is a large percentage of the total fuel cost. In large stations the amount of fuel and ash handled frequently warrants the expense of elaborate conveyor systems which would not be justified in smaller plants. In whatever way coal is supplied provision should be made for storing a quantity sufficient to operate the plant for some time in case the supply is interrupted, thereby guarding against an enforced shut-down.

If adjacent to a railway line, a side track must be provided for switching the cars. As bottom-dumping cars cannot be depended upon, provision should be made for unloading by hand. If coal is delivered by water, clam-shell drop buckets are ordinarily used for unloading the barges. If the power house is located at some distance from the railroad or water the coal is generally hauled by teams in two to five-ton loads.

115. Coal Storage. — In small stations the storage bins or coal bunkers may usually be located within the building, but in larger plants the quantity of coal consumed daily is frequently such that an immense space would be required to furnish storage capacity for even a short period of time. For example, one of the large central stations in Chicago burns an average of 30 tons of Illinois screenings per hour throughout the year. Allowing 45 cubic feet to the ton this would necessitate a space of 45 x 30 x 24 = 34,800 cubic feet to store coal for one day's operation. A ten-days run would require a coal pile 50 feet wide, 30 feet high, and 232 feet long. It is a good plan, if the location and character of the plant permit, to carry four or five days' supply within the plant and provide a separate building for the coal reserve. Such provision is made in the power plant of the New York Edison Company, which has a storage capacity of 150,000 tons in addition to that of the overhead bunkers.

Exposed coal piles are objectionable, because of freezing in winter, the crust sometimes freezing so hard as to necessitate the use of dynamite to break it; moreover, a slow depreciation in heat value takes place, especially with bituminous coal. This depreciation is more rapid in warm weather and in the tropics. Stored coal is oftentimes subject to spontaneous combustion, particularly when there is a large content of iron pyrites.

Coal bunkers or hoppers are ordinarily placed on the same level with boiler-room floor or above the boiler setting. The former is the cheaper as far as first cost is concerned, but necessitates additional handling of the fuel before it can be fed to the stokers. In the overhead system the coal gravitates to the stoker through down spouts. Overhead bunkers are usually found where real estate is costly. They are generally constructed of steel plates lined with concrete or reënforced concrete. The bottoms slope at an angle of 35 to 45 degrees and empty into the coal chutes or down spouts. Fig. 99 shows the general appearance of a single overhead bunker and Fig. 441 that of a double bunker. In some bunkers the floors are made with very slight slopes, but it is not advisable to use a slope less than the angle of repose of the coal, as it may be necessary to shovel the coal over the spouts. Convenience in framing makes the 45-degree slope the more desirable. Separate bunkers for each boiler are preferred to continuous bunkers, since fire in the coal is more readily prevented from spreading. In the new power house of Swift & Co., Chicago, Ill., the bunkers are of circular cross section instead of rectangular as is the usual practice. The capacity of the cylindrical hopper is considerably less than that of a rectangular hopper of the same width, but is much cheaper to construct.

Ash bins are invariably lined with concrete or brickwork, since the corrosive action of the ashes would soon destroy the bare iron, and are usually located alongside the coal hopper, as in Figs. 97 and 99, so that they may be discharged by gravity. The angle of repose of most ashes is approximately 40 degrees, but the 45-degree angle is preferred on account of convenience in construction.

Coal Storage: Power, April, 1907, p. 217, Aug., 1899, p. 3, Nov., 1904, p. 651; Eng. Rec., Sept. 23, 1905, p. 534, June 1, 1902, p. 532, July 4, 1903, p. 4; Eng. News, July 11, 1907, June 5, 1902, p. 463, April 3, 1903, p. 272; West. Elec., Oct. 28, 1905, p. 335; Trans. A.S.M.E., 23–473.

Coal Storage under Water: Eng. News, Dec. 24, 1908; Eng. Min. Jour., Dec. 1, 1904, p. 975; Engineering, Sept. 4, 1903, p. 863.

Calorific Value of Weathered Coals: Trans. A.S.M.E., 20–333; Bulletin No. 17, Univ. of Ill., Aug. 26, 1907.

Design of Coal and Ash Bins: Eng. News, July 21, 1904, p. 62; Eng. Rec., Sept. 1, 1900, p. 201; Power, Nov., 1899, p. 14, Nov. 1904, p. 651; Elec. Age, March, 1907, p. 141.

116. Coal Conveyors. — Coal is carried to the stokers in a variety of ways, depending upon the location of the plant, the type of stokers, and the personal tastes of the builder. Of the various methods the following are the most common :

1. Hand shoveling from coal pile to furnace.
2. Wheelbarrow or hand car and shovel.
3. Bucket conveyor.
4. Belt conveyors.
5. Hoist and hand cars.
6. Hoist and automatic cable cars.
7. Combinations of the above.

Coal-Handling Plants for Power Houses: Am. Elec., June, 1900, p. 266, Oct., 1901, p. 486; Cassier's, April, 1905, p. 480; Elec. World., Dec., 1901, p. 463; Engr., U.S., July 1, 1904, p. 461; Jan. 1, 1905, p. 4; Eng. Rec., April 5, 1902, p. 322.

Ash Handling: Elec. World., Oct. 5, 1901, p. 569; Eng. News, Oct. 19, 1905, p. 403; Eng. Rec., May 10, 1902, p. 435, Jan. 17, 1903, p. 85, Feb. 7, 1903, p. 153, Oct. 28, 1905, p. 482, Oct. 7, 1905, p. 396, Dec. 9, 1905, p. 655; Power, Oct., 1904, p. 507, July, 1904, p. 422; St. Ry. Jour., Jan. 5, 1901, p. 11.

117. Hand Shoveling. — Where possible the coal is dumped direct from the cars or wagons into bins located in front of the boilers. In such instances one man may handle the coal and ashes and attend to the water level of 200 horse power of boilers equipped with common hand-fired furnaces. With stoking and dumping grates 300 horse power may be controlled by one man and from 800 to 1000 horse power with chain-grate stokers. This refers of course to average good coal not too high in ash nor productive of much clinker. Sometimes the coal cannot be stored in front of the boilers but must be hauled by wheelbarrow, cart, or rail car. For distances over 100 feet and quantities over 20 tons per day the cost of handling the coal in this way may justify the installation of an automatic conveyor system. Hand-fired furnaces and manual handling of coal and ashes are usually associated with small plants of 500 horse power and under, but a number of large stations are operated in this way with apparent economy. A notable example is the new (1907) steam-power plant of the Wood Worsted Mill, Lawrence, Mass., in which 40 return tubular boilers are fired by hand. A tipcart with a capacity of one ton brings the coal a distance of 100 to 200 feet to the firing floor, and firemen shovel it on to the grate. Four men are stationed at the coal pile. One man drives two carts (one of which is being filled while the other is gone with its load), sixteen firemen attend to the furnaces, and two men dispose of the ashes.

Most large plants, however, are equipped with conveying machinery, not so much because of the possible reduction in cost of operation, taking into consideration all charges fixed and operating, as because of the large and often unreliable labor staff which it dispenses with. Hand shoveling is sometimes necessary even with modern unloading devices on account of the freezing of coal in the cars. This is particularly true of washed coals, and it is not unusual to have an entire car load solidly frozen. In this case it has to be picked and shoveled by hand, or the unloading tracks must be equipped with steam pipes and outfits for thawing purposes. A good man is capable of shoveling 40 to 50 tons of coal in eight hours when unloading a car, provided it is only necessary to shovel the coal overboard.

118. Bucket Conveyors. — One of the most common methods of automatically handling the coal from car to bunker is by means of an endless chain of traveling buckets. Many of the largest central stations in this country are equipped with such systems. The details of operation are best illustrated by a few examples.

Fig. 96 gives a diagrammatic arrangement of the link-belt overlapping pivoted bucket carrier, and Fig. 97 illustrates its application to a typical boiler plant. Coal is discharged from the railway cars into a track hopper and from there delivered by a "feeding apron" into a crusher which reduces it to such a size as can be conveniently handled by the stokers. It is then discharged into a short bucket conveyor, which carries it to the main system of buckets, and it is elevated to the proper level and discharged into the overhead bunkers. The discharge is effected by special tripping devices which engage the buckets and turn them over. The ashes are dumped from the ash pit through a series of chutes into the lower run of buckets, by which they are elevated and discharged into the ash hopper alongside the coal bunkers. From the ash hopper the ashes discharge by gravity directly into the railway cars below. The system is operated by means of two motors, one driving the crusher and the other the main bucket system. The buckets are made of either sheet steel or malleable iron.

In Fig. 96 the coal is fed to the crusher by the "reciprocating feeder," which is usually placed directly under the track hopper. The feeder consists of a heavy steel plate mounted on rollers and having a reciprocating movement effected by a crank mechanism from the carrier. The amount of coal delivered depends upon the distance the plate moves, and this can be varied by changing the throw of the eccentric. The number of strokes corresponds to the number of buckets. Any size coal can be readily handled. When the distance from track hopper to carrier is so great that the reciprocating feeder is not practicable a

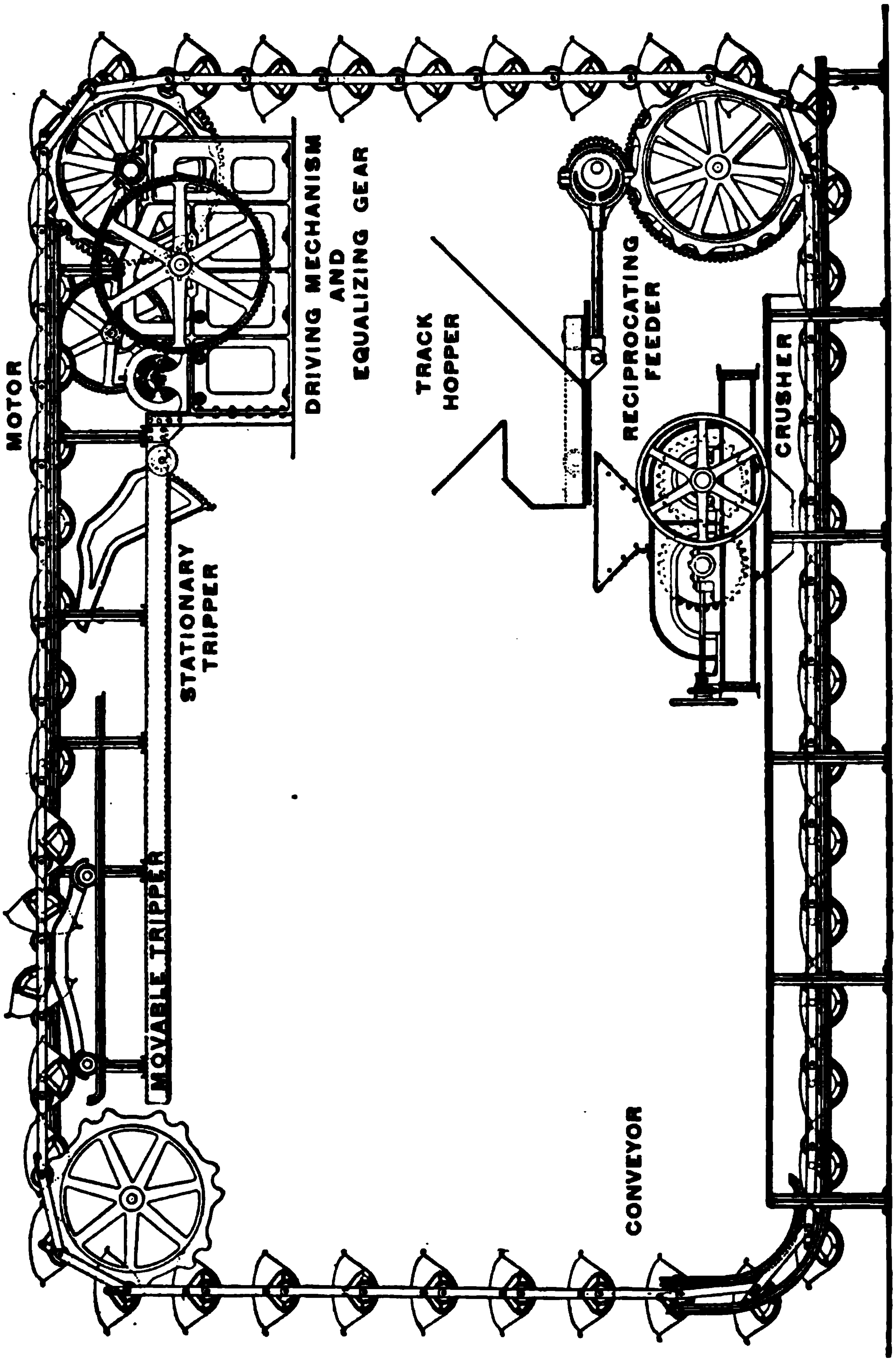


FIG. 96. Link-belt Coal-Handling Apparatus.

Fig. 97. Typical Coal and Ash-Handling Equipment; Bucket Conveyors.

FIG. 98. Steel Cable Company's Coal-Handling Apparatus.

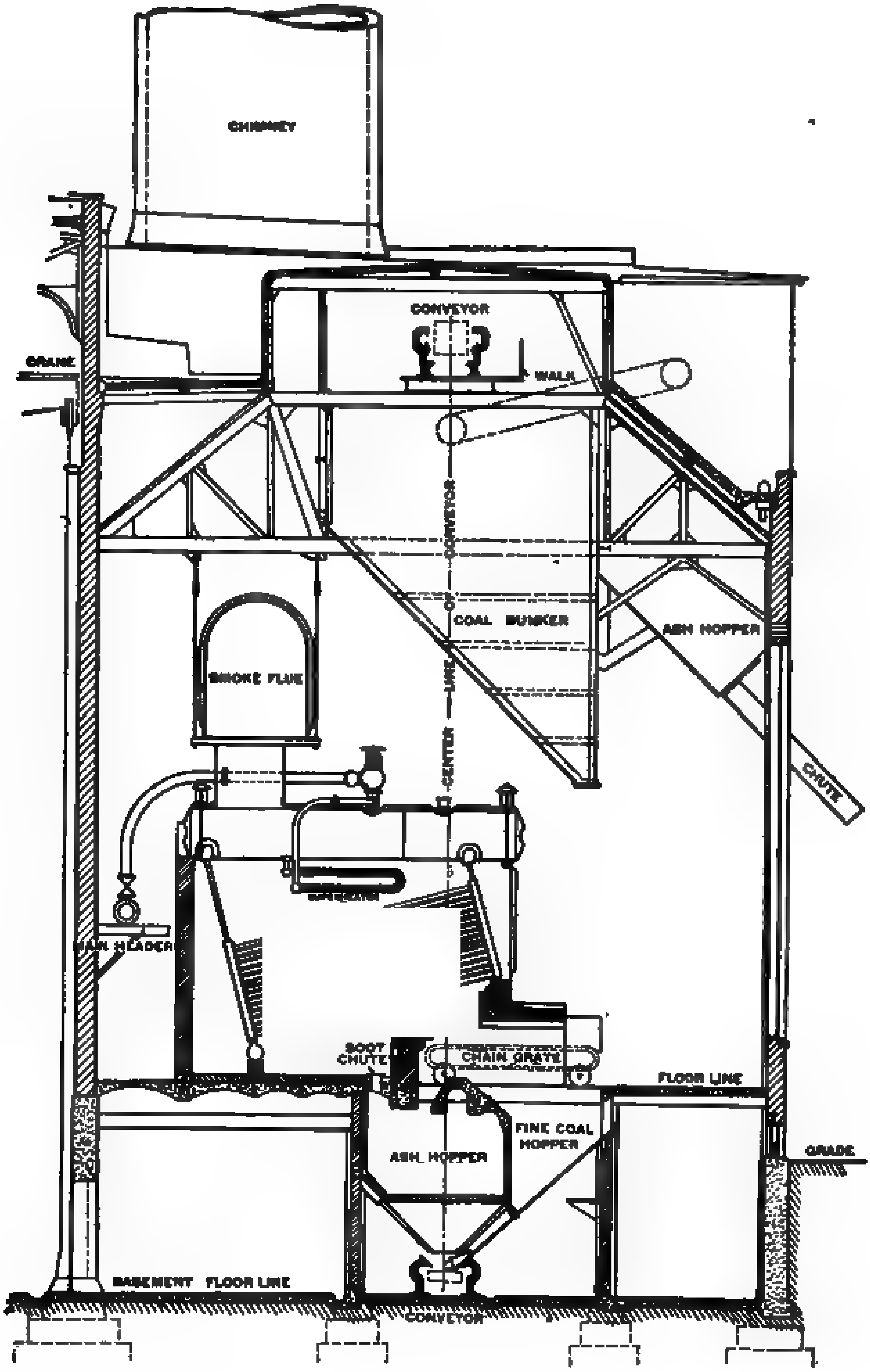


FIG. 99. Coal and Ash-Handling System in the Power House of the South Side Elevated Railway Company, Chicago.

continuous or "belt" feeder is used to supply the crusher with fuel. The "equalizing gear" is designed to impart a pulsating motion to the driving sprocket wheel which will counteract the natural pulsation to which long pitch chains are subject, producing violent increase of the normal strain at frequent intervals. This is accomplished by driving the spur wheel with an eccentric pinion, causing the pitch line to describe a series of undulations corresponding to the number of sprockets on the chain wheel. Figs. 99 and 100 show the general arrangement of crusher and "cross conveyor" in the old portion of the South Side Elevated Power House, Chicago.

A coal and ash system similar to the one illustrated in Fig. 97 for a plant consisting of eight 350-horse-power boilers will cost in the neighborhood of $8,000, completely installed. This does not include the cost of coal and ash bunkers.

The Hunt conveyor, Fig. 101, while usually called a "bucket" conveyor, is in fact a series of cars connected by a chain, each having a body hung on pivots and kept in an upright position by gravity. The chain is driven by pawls instead of by sprocket wheels. The "buckets" are upright in all positions of the chain, consequently the chain can be driven in any direction. The change of direction of the chain is accomplished by guiding the carriers over curved tracks. The chain moves slowly, and the capacity is governed by the size of the buckets. The ordinary size buckets carry two cubic feet of coal and move at a rate of fifteen buckets a minute, carrying about 40 tons per hour. Two methods of filling the buckets are employed, the "measuring" and the "spout filler." In the former each bucket is separately filled with a predetermined amount by a suitable "measuring feeder." In the latter the material is spouted in a continuous stream, necessitating the use of overlapping buckets to prevent spilling of the material. Fig. 102 shows an application of the Hunt system to the old plant of the Baltimore United Railways and Electric Company.

Fig. 103 gives a sectional elevation of the coal and ash-handling machinery at the power plant of the Commercial National Bank Building, Chicago. Underneath the sidewalk on the Clark Street side of the building is a coal-storage bin of 600 tons capacity, served with a bucket conveyor. One leg of the conveyor reaches down to a level below the track of the Illinois Tunnel Company. By this arrangement coal can be delivered either by cars in the tunnel or by wagons from the street. In taking coal from storage a gate at the lower extremity of the hopper is opened and the coal filling the buckets is elevated and tripped into any one of the screw conveyors leading from bucket conveyor to boiler hopper. The ashes are shoveled from the ash pits into

Fig. 100. Crusher and Conveyor of S. S. Elevated Railway.

FIG. 101. Driving Mechanism of Hunt Conveyor.

FIG. 102. Coal and Ash-Handling System at the Old Power House of the Baltimore United Railways and Electric Company.

cars running in a cross tunnel under the boiler floor, and by these cars are transferred to a dump at one side of the boiler room and discharged into Illinois Tunnel Company's cars for removal.

FIG. 103. Bucket and Screw Conveyor at Commercial National Bank Building, Chicago, Illinois.

119. Belt Conveyors. — The Robins belt conveyor, Fig. 104, consists essentially of a thick belt of the required width driven by suitable pulleys and carried upon idlers so arranged that the belt becomes trough-shaped in cross section. The belt is constructed of woven cotton duck covered with a special compound on the carrying side. The belt is thicker at the middle than at the edges, since the wear is greatest in a line along the center. The idlers are carried by iron or wooden framework, and are spaced from 3 feet to 6 feet between centers on the troughing side according to the width of belt and the weight of the load. On the return side these distances range from 8 to 12 feet.

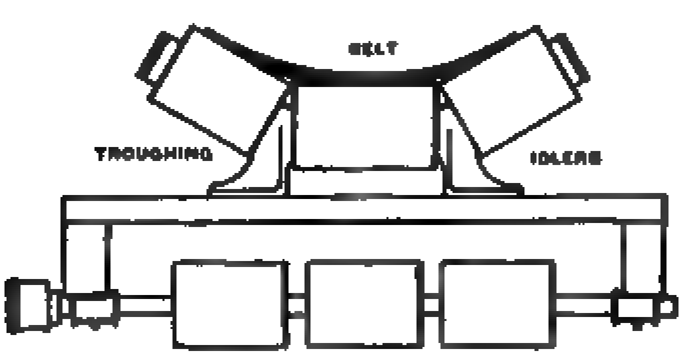


FIG. 104. Guide Pulleys, Robins Belt Conveyor.

High-speed rotary brushes with interchangeable steel bristles prevent wet, sticky material from clinging to the belt. Automatic tripping devices placed at the proper points cause the material to be discharged where it is needed. The trippers consist essentially of two pulleys, one above and slightly in advance of the other, the belt running over the upper and under the lower one, the course of the belt resembling the letter S. The material is discharged into chutes on the first downward turn of the belt. The trippers may be movable or fixed, single or in series. Movable trippers are used when it is desired to discharge the load evenly along the entire length, as, for instance, in a continuous row of bins, while fixed trippers are employed where the load is to be discharged at certain and somewhat separated points. The movable trippers are made in two forms, "hand-driven" and "automatic." In the former they are moved from point to point by means of a hand crank. The "automatic" tripper is propelled by the conveying belt through the medium of gearing. It reverses its direction automatically at either end of the run, and travels back and forth continuously distributing its load. It can be stopped, reversed, or made stationary at will. The most notable installations of this system are at the 96th Street station and the Kingsbridge station of the Metropolitan Street Railway Company, New York City.

120. Elevating Tower, Hand-Car Distribution. — Fig. 105 illustrates the coal and ash-handling installation at the Aurora and Elgin Interurban Railroad power house, Batavia, Ill. Coal is delivered to the plant by railroad cars which dump directly into coal hoppers located inside a steel structure running the entire length of the building and spanned by two railroad tracks. There are 18 hoppers constructed of 17-inch brick walls fitted with steel-plate bottoms. Subdividing the storage space in this manner makes it possible to carry different grades of coal, prevents the spreading of fire, and affords a simple construction for the support of the railroad tracks. The basement of the boiler room extends underneath the hoppers, and two lines of narrow-gauge tracks are imbedded in the concrete floor. Turntables at the center facilitate the switching of cars to the elevators which rise through the boiler room close to the chimney. The cars, of one ton capacity each, are of special construction, with roller-bearing axles and a combined ratchet lift and friction dump. The filled cars are pushed from underneath the hoppers to two elevators which lift them to the line of tracks supported overhead across the boiler fronts. They are then pushed to the hoppers suspended above the boiler setting and the coal is dumped. These hoppers have a capacity of six tons each. From the hoppers the coal is fed to the stoker by an ordinary down spout.

The ashes fall from the stokers into an ash pit, from which they may be discharged into ash cars. The ash cars are elevated to a set of tracks running at right angles to the main tracks, and are transferred to ash bins located directly over the coal bins. Coal and ashes are weighed

FIG. 105. Coal and Ash-Handling System at the Power House of the Aurora and Elgin Interurban Railway, Batavia, Ill.

in the small cars. There are ten boilers in this plant and four men are required to handle the coal and ashes. The entire coal and ash-handling system cost about $10,000, and the cost of handling the coal and ashes is approximately 4 cents per ton. This does not include wages of firemen or water tenders.

121. Overhead Storage, Bucket Hoist. — Fig. 106 gives a general view of the coal-handling plant of the Depot Street power house of the Cincinnati Traction Company. This installation is a good example of an application of the "overhead storage gravity feed" system to an existing plant without interfering in any way with its operation. The system consists essentially of a receiving pit below the car tracks from which the coal is hoisted to a series of overhead bins. The coal storage is outside the boiler house in an independent structure. The bins are of steel framework with concrete floors, and are sufficiently elevated

Fig. 106. Coal and Ash-Handling System at the Depot Street Power House of the Cincinnati Traction Company.

to spout coal easily to the stoker magazine. The total capacity of the overhead bins is about 1,600 tons. The four bins or receiving pits have a capacity of 50 tons each, or approximately one car load, and are so situated that all four may be filled simultaneously without shifting the train. The coal-handling apparatus consists of a one-ton self-filling bucket operated on a three-motor electric crane running on rails at the top of the storage bins. The coal is hoisted from the receiving pit through suitable shafts in the bin structure and dumped into the overhead hoppers. The maximum capacity of the hoist is 50 tons per hour. The labor required to handle the coal from car to bins is performed by one man working five hours per day and an assistant engaged a small part of the time to dump cars, clean hoppers, etc. The average daily coal consumption is approximately 200 tons. The total cost of the

equipment was about $18,000 for the bins complete and $4,500 for the coal-handling crane. The cost of handling the coal and ashes is approximately 1.5 cents per ton of coal. Including all charges fixed and operating the total cost of handling the coal is about 3.5 cents per ton. This does not include wages of firemen or water tenders.

122. Elevating Tower, Cable-Car Distribution. — The coal and ash-handling system of the new turbine power plant of the Detroit Edison Company, Fig. 107, is a typical example of a large station equipped with elevating tower and cable-car distributers instead of the usual bucket conveyor. The system consists essentially of a lofty steel tower in which are housed at various levels a track receiving hopper, crushing rolls and screens, weighing hopper, hoisting apparatus, etc., and a small cable railway for delivery to the bunkers. The railroad coal cars enter the tower on an elevated trestle 18 feet above grade, below which is a track receiving hopper. A two-ton "tub hoist" is filled with coal from the bottom of the receiving hopper and elevated to a 20-ton bin at the top, 120 feet above ground level. This bin has a grille bottom at one side and under the outlet a heavy duty coal crusher, thus allowing the fine coal to screen through directly while all the larger lumps are automatically delivered to the crusher. The hopper beneath this delivers to the revolving screen, which sorts the slack into one bin below and the nut coal into the other. From the two bins the small cable cars are filled for dumping into the desired bunkers over the boiler rooms. The cars are arranged for automatic dumping by means of adjustable trips which may be located at any point. The object of separating the nut coal and slack is to burn the latter during light or medium loads, keeping the former for heavy loads and "peak" overloads. The down spouts are double, with a valve in each branch operated from the floor, so that either grade of fuel may be drawn out at any time and in any proportion desired. The entire system has a capacity of from 50 to 75 tons of coal per hour and is driven by steam engines, with the exception of the revolving screen which is motor driven. The ash-handling system consists of brick-lined concrete hoppers underneath each pair of stokers which discharge their contents by gravity into the small cars operated on the track system in the boiler-house basement.

When handling 275 tons per day of 24 hours the cost of operation is approximately 12.5 cents per ton from coal car to ash car, including wages of firemen and water tenders.

123. "Vacuum" Ash Conveyor. — Fig. 108 gives a diagrammatic arrangement of a recently patented ash-conveying system depending upon the velocity of a column of air for moving the ashes. The system

Fig. 107. Coal and Ash-Handling System at the Power House of the Detroit Edison Company.

is simple in operation and low in first cost. One end of special cast-iron header *F* leads to the ash pits of the various boilers by means of branch tubes, and the other end is connected with a sealed separating chamber *A*. Each branch pipe is fitted with simple circular openings directly underneath each ash-pit door for admitting ashes and which are kept covered except when in operation. Exhauster *E* creates a partial vacuum in chamber *A* and draws in air at a high velocity from the opening in the ends of the branch pipes. Ashes raked into the

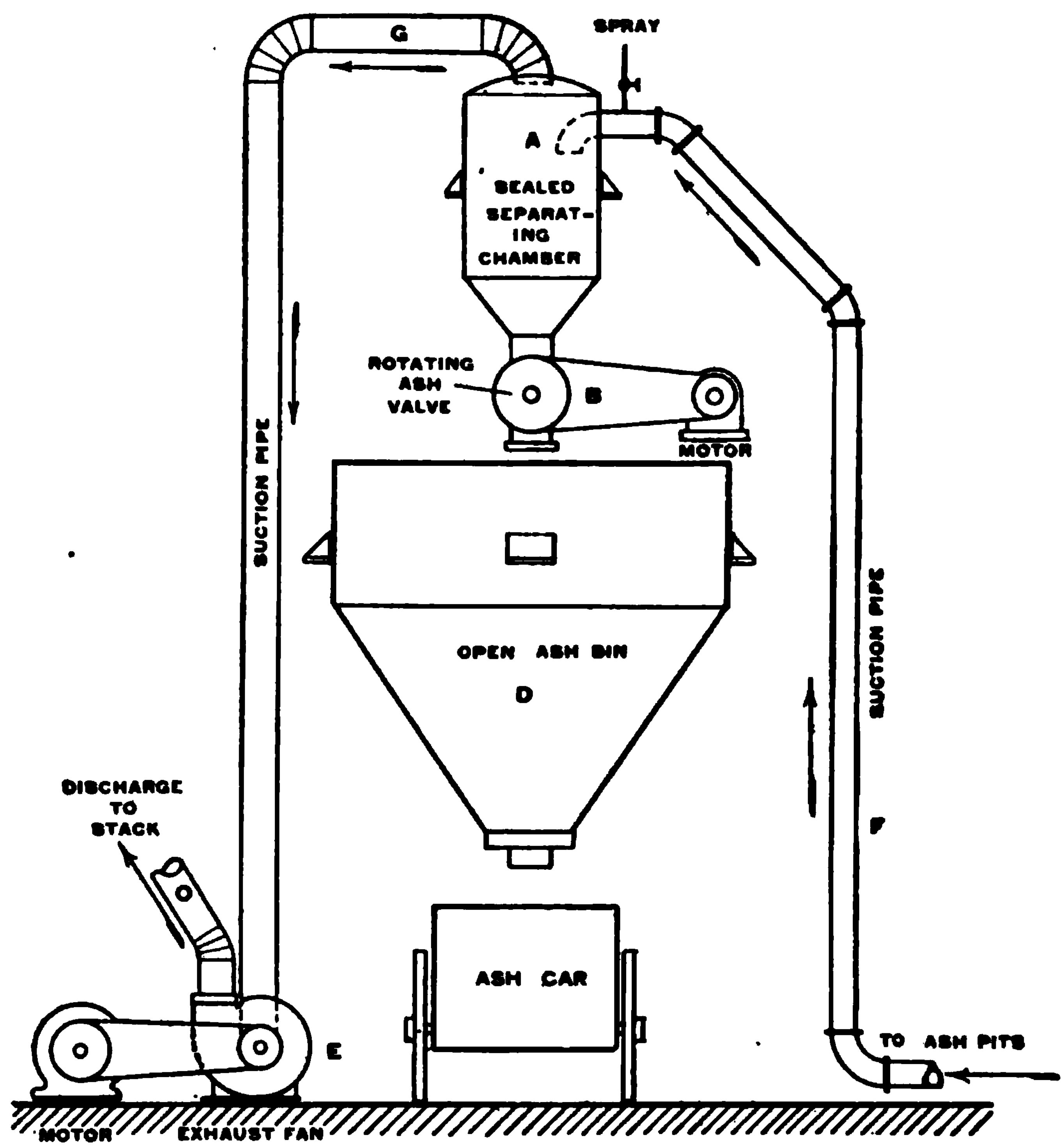


FIG. 108. Diagrammatic Arrangement of the "Vacuum" Ash-Handling System.

pipes through the openings are caught by the rapidly moving column of air and forced into chamber *A*. The ashes fall to the bottom and are fed into the main ash pit by a slowly revolving ash valve *B*. Air and dust are withdrawn from the top of the separator chamber through pipe *G* and discharged to the stack or to waste. A spray is introduced into pipe *F* to reduce dust. The process is a continuous one, and the ashes may be completely removed from the ash bin without interfering with the operation of the exhauster. In a later construction the ash

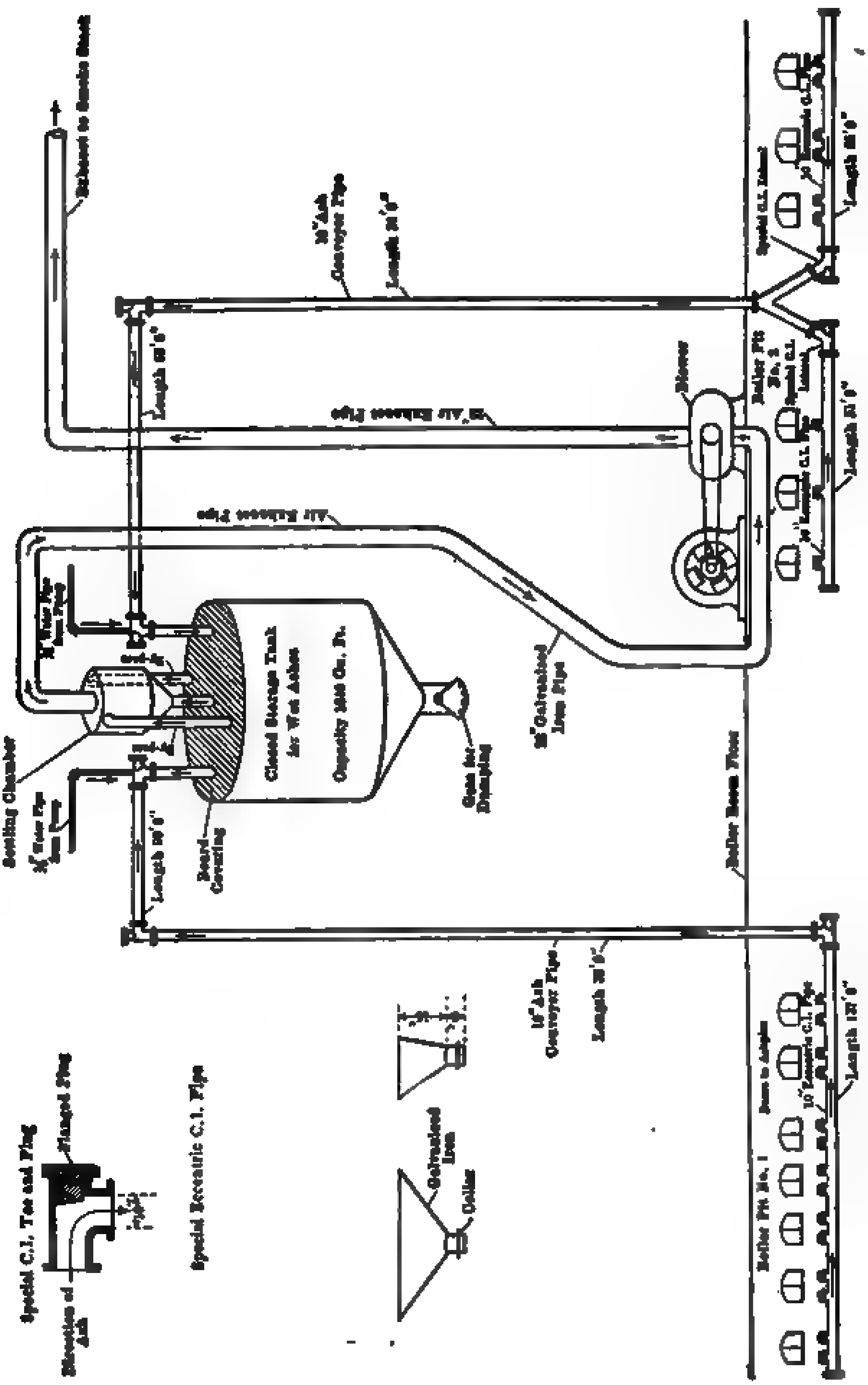


FIG. 108a. Diagrammatic Arrangement of Vacuum Ash-Handling System at the Armour Glue Works, Chicago, Ill.

FIG. 108b. Coal and Ash-Handling System. Norfolk Traction Co., Norfolk, Va.

bin and separating chamber are included in one chamber, thus doing away with the revolving ash valve and the small motor operating it. In this latter design the bin is never completely empty, a certain depth of ashes being maintained to seal the bottom at all times.

At the Armour Glue Works, Chicago, Ill., this system is applied to a boiler plant of thirteen boilers, aggregating 4,800 horse power, and cost, completely installed, $5,600. As originally installed the separating chamber had a volume of about 35 cubic feet and the suction intake was placed 58 feet above the ash-pit level. The revolving ash valve made about 13 r.p.m., and was driven by a one-horse-power motor. In the present installation the separating chamber and motor-operated ash valve are dispensed with and the discharge pipes lead directly into the main ash bin, which has a capacity of 60,000 pounds of wet ashes and is constructed of five-sixteenths-inch sheet iron. The exhauster (a 30-foot Root blower) has a capacity of about 8,000 cubic feet per minute at 265 r.p.m., and is driven by a 75-horse-power motor. Under normal conditions of operation the motor requires 50 horse power when delivering 250 pounds of ash per minute, and the vacuum on the suction side of the exhauster is 3.3 inches of mercury. The pipe from the ash bins to the separating chamber is 10 inches in diameter and is constructed of extra heavy chilled cast-iron pipe. The piping from the separating chamber to exhauster and to stack is 22 inches in diameter and is constructed of number 16 and number 20 galvanized iron. The ashes are raked by hand from the ash pits to the suction openings of the branch pipes, and are handled dry, the dust being taken along with the ashes. Elbows are soon worn out by the abrasive action of the ashes, and tees are used instead, since the accumulation in the " dead " end receives the impact and takes up the wear. The cost of handling the ashes in this installation is approximately 7 cents per ton.

124. Cost of Handling Coal and Ashes. — In large stations where a number of men are employed to handle coal and ashes only it is a simple matter to divide the cost of handling into the various stages, thus :

1. Cost of unloading cars or barges.
2. Cost of conveying coal to bunkers.
3. Cost of feeding coal to furnace.
4. Cost of removing ashes.

These costs are usually expressed in cents or dollars per ton of coal burned, or in terms of cents or dollars per horse power hour or kilowatt hour of main prime mover output. Item number 3 is oftentimes included under "boiler-room attendance" and items 1, 3, and 4 under " coal and ash handling." Not infrequently all four items are included under " attendance." So much depends upon the character of stokers

and furnace, size of boilers, and the like, that general figures on the cost of handling the coal and ashes are of little value unless accompanied by a description of the equipment. For the sake of general comparison the most satisfactory method of expressing the cost is in dollars per ton of coal from coal car to ash car. This includes wages of coal and ash passers, repair men, and boiler tenders. In small stations the coal and ash handling is done by the boiler tenders, in which case it is impracticable to separate the items mentioned above, and the cost is ordinarily included under attendance. An average figure for handling coal by barrow and shovel is not far from 1.6 cents per ton per yard up to the distance of five yards, then about 0.1 cent per ton per yard for each additional yard. With automatic conveyors the operating cost, not including wages of firemen and water tenders, varies with the size of plant and the type of conveyor, and ranges anywhere from a fraction of a cent per ton to four or five cents per ton. The larger the plant and the greater the amount of coal handled the lower will be the cost per ton. In comparing the relative costs of manual and automatic handling, fixed charges of at least 15 per cent of the first cost of the mechanical equipment should be charged against the latter in addition to the cost of operation. In large central stations equipped with stokers and conveyors and consuming 200 tons or more of coal in twenty-four hours, the cost of handling the coal from coal car to ash car, including wages of firemen and water tenders, will range between 10 cents and 18 cents a ton.

125. Coal Hoppers. — Fig. 109 shows a front and side elevation of a typical set of stationary weighing hoppers as applied to the boilers of the Quincy Point power plant of the Old Colony Street Railway Company, Quincy Point, Mass. Each battery of boilers is provided with an independent set of hoppers. The bottoms of the overhead coal bunkers lead into the small hoppers *A*, *A*. The operation of any single weighing hopper is as follows: Coal is fed from the overhead bunkers to weighing hopper *H* by means of valve *V*. The weight of coal in the weighing hopper is transmitted by a system of levers and knife edges to the inclosed scale beam *I* and noted in the usual way. The weighed charge of coal is then admitted to the down spout *S* by means of valves similar to those at *V*.

Although separate weighing hoppers for each battery, as illustrated in Fig. 109, offer many advantages, they are quite costly and it is not unusual to install one or more large weighing hoppers mounted on overhead traveling carriages so that one may supply a number of boilers (Fig. 110). At the Armour Glue Works, Chicago, the coal supply is stored in one large overhead bunker of 1000 tons capacity. A five-

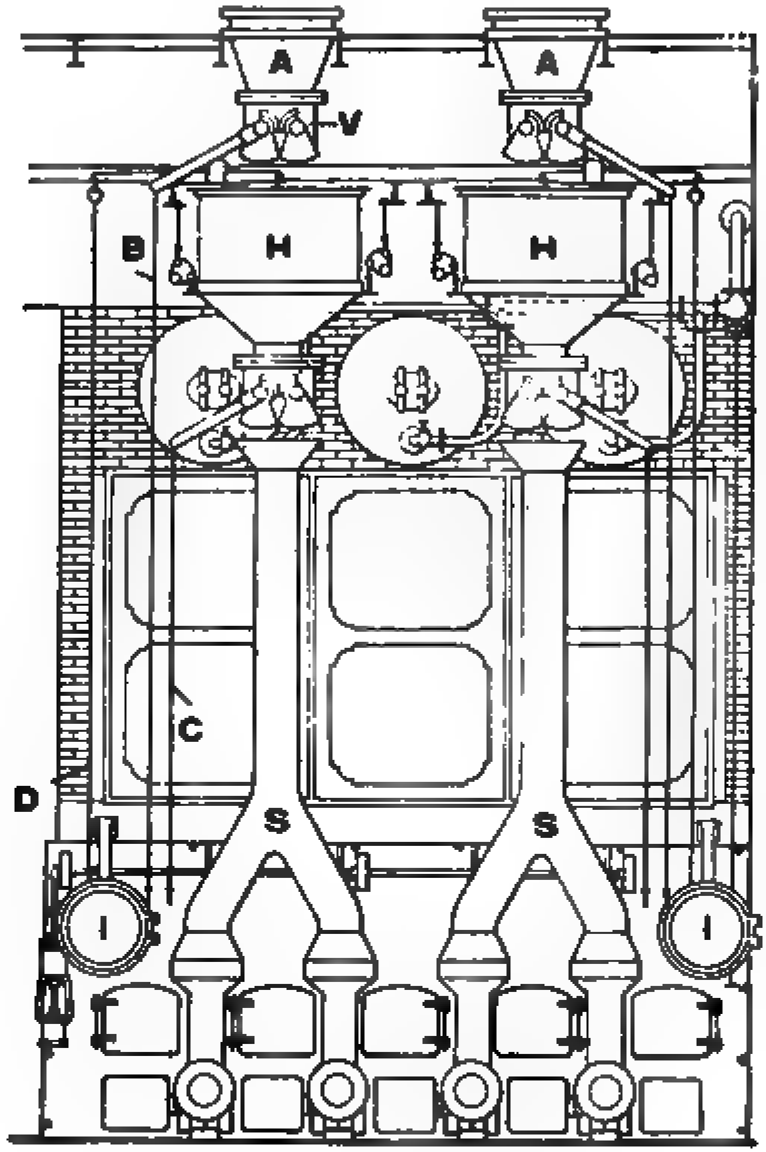


FIG. 109. Stationary Coal Weighing Hoppers.

FIG. 110. Traveling Coal Hoppers.

ton motor-driven traveling hopper receives its supply from this central bunker and delivers it to the various boilers. One man operates the

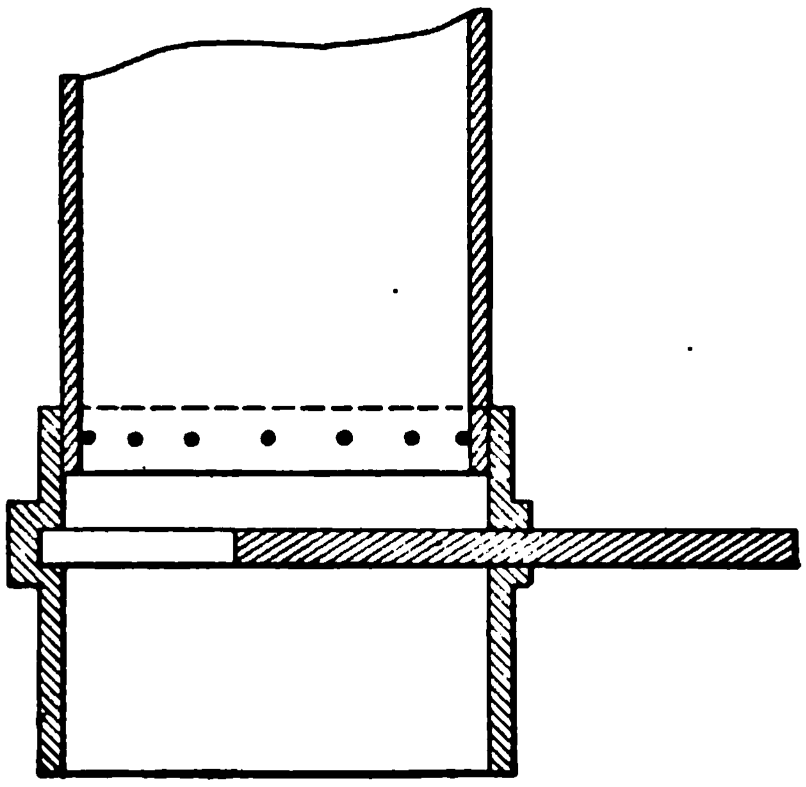


FIG. 111. Common Slide Coal Valve.

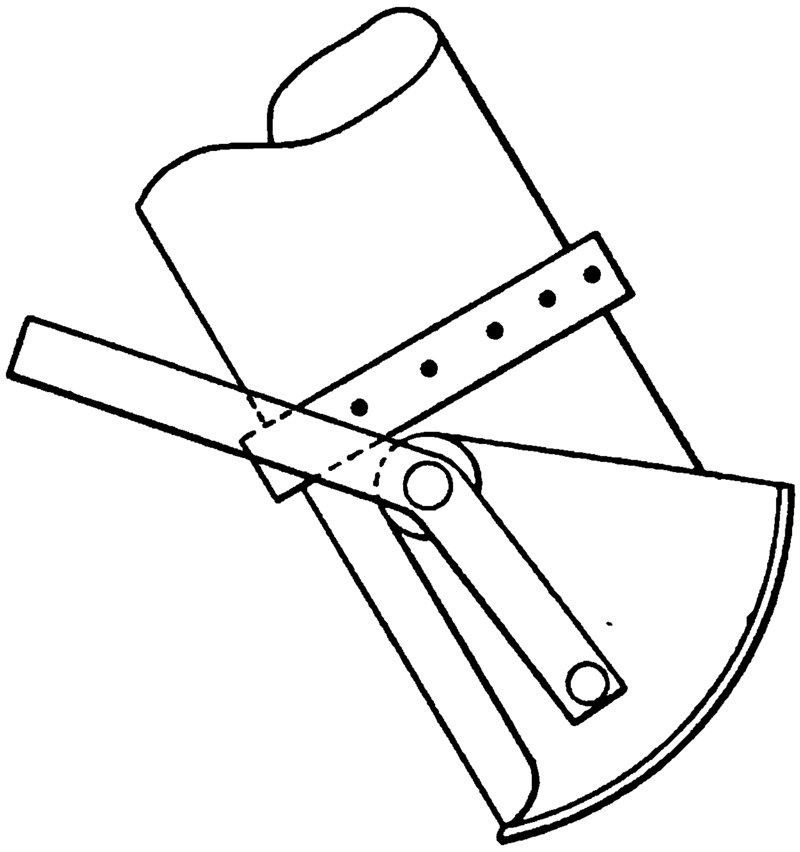


FIG. 112. Simplex Coal Valve.

traveling hopper, tends to the coal valves, and supplies all boilers with coal.

Weighing hoppers are sometimes made automatic; that is, the opening and closing of valves, feeding of coal, and recording of weight are automatically performed by the weight of the coal itself. The scale is set for discharges of a certain weight and continues to discharge this amount automatically. In the few plants which are equipped with automatic weighing hoppers the capacity of the hopper is approximately 100 pounds per discharge. These hoppers are necessarily more complicated and more costly than the ordinary weighing hoppers, and it is a question whether the advantages offset the extra first cost and maintenance charges. A small automatic hopper of 100 pounds discharge capacity costs approximately $400 as against $250 for the ordinary weighing device.

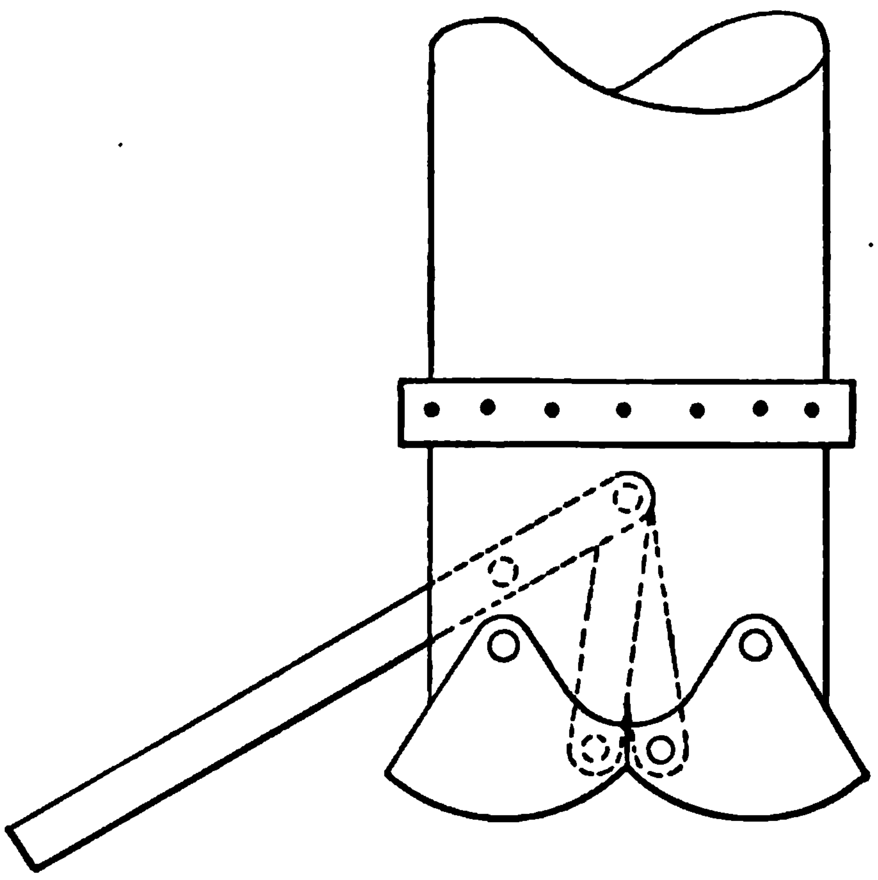


FIG. 113. Duplex Coal Valve.

126. Coal Valves. — Figs. 111 to 115 illustrate the principles of a few well-known coal valves. They may be conveniently grouped into two classes according to the location of the coal pocket: (1) those drawing the coal from overhead bunkers and (2) those drawing from the side of a bin. In the first class come the simple *slide valve*, the *simplex* and *duplex rotating* valve. In the latter are the *flap* valve and the *rotating* valve. They are made in various sizes and designs, but those illustrated are examples of the most common types. The simple slide valve, Fig. 111, is applicable only to small size coal and to small spouts, since coarse or lump coal may get in the way and prevent proper closing. The simplex valve, Fig. 112, consists of a rotating jaw actuated by a lever. There are no rubbing surfaces, and the jaws cut through the material without jamming. The duplex valve, Fig. 113, consists of two rotating jaws connected to a common actuating lever. The jaws move simultaneously, so that even a partially open valve delivers the coal centrally. When closing the valve the flow is gradually stopped by the decreasing width of the opening and there is but little resistance to the movement of the jaws. The largest valve can easily be operated by hand.

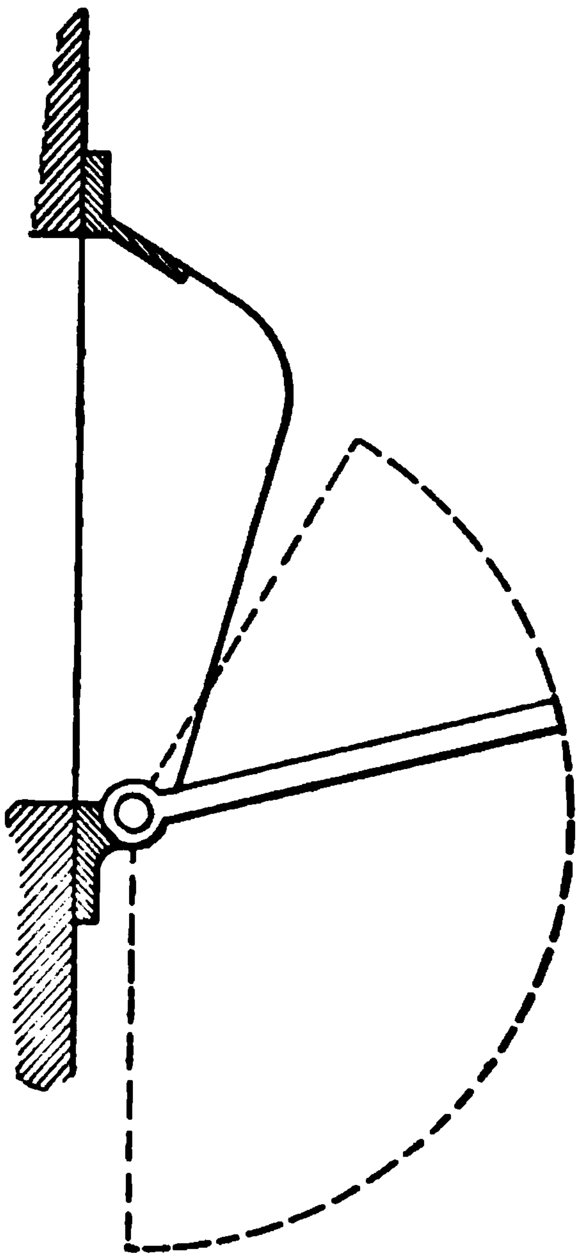


Fig. 114. Common "Flap" Coal Valve.

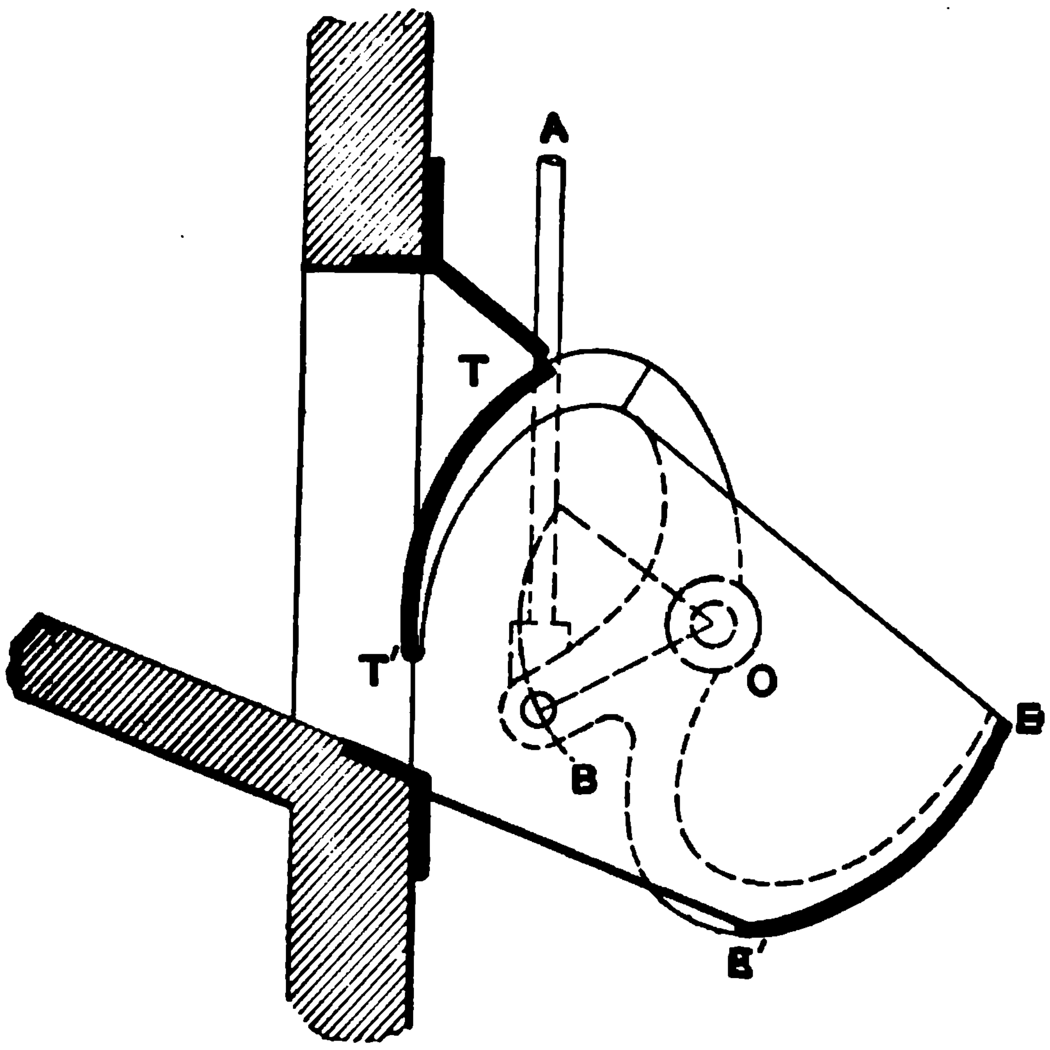


Fig. 115. "Seaton" Coal Valve.

The flap valve, Fig. 114, is the simplest form for drawing coal from a side bin. It consists merely of an iron flap hinged to the bottom of

the chute. The valve is lowered to let the coal run over its top and is raised to stop the flow. It cannot be clogged or get jammed in closing. The flap is raised and lowered by a simple lever. For very large bins, where the valves are to be opened and closed frequently, the "Seaton" valve, Fig. 115, is usually preferred. This valve consists of two jaws EE', and TT' pivoted to suitable framework at O and actuated by lever A. The valve is shown fully closed. Raising lever A causes the cut-off blade EE' to rotate about O and permits the coal to flow through the space between the edge of the jaw E and the end of the chute. The rate of flow is regulated by the width of this opening. The cut-off blade does not reach a stop, hence there is no possibility of a lump of coal getting in the way and preventing the prompt closing of the valve.

Coal and Ash-Handling Installations: Commonwealth Edison, Chicago, Power, Dec., 1906, p. 718. Boston Elevated, Elec. World, Sept. 7, 1901, p. 396. Interborough Rapid Transit Co., New York, Elec. World, Feb. 4, 1905, p. 264; Engr., U.S., May 15, 1904, p. 337; Eng. News, Jan. 14, 1904, p. 41. Waterside Station, New York, Edison Co., Eng. Rec., Sept. 9, 1905, p. 287. Detroit Edison Co., Eng. Rec., Oct., 1905, p. 396. Brooklyn Rapid Transit Co., St. Ry. Jour., Sept. 23, 1905, p. 435. N. Y. C. and H. R. R., St. Ry. Jour., Nov. 11, 1905, p. 876. Aurora, Elgin and Chicago Ry., Eng. Rec., Feb. 7, 1903, p. 153. Missouri River Power Co., Eng. News, Oct. 19, 1905, p. 403. Brooklyn Edison Co., Gold St. Station, Elec. World, June 15, 1907.

Hoisting and Conveying Machinery: Pro. A.S.M.E., June, 1908.

CHAPTER VII.

CHIMNEYS.

127. Chimney Draft.—Draft produced by a chimney depends upon so many conditions and involves such a large number of variables that empirical methods of proportioning, based upon actual performances, are more to be relied upon than theoretical calculations. Draft is due to the difference in the weight of the column of hot light gases in the stack and that of the cooler and heavier surrounding atmosphere, the latter tending to flow into the base and thereby force the lighter gases out the top of the stack. The commonly accepted theory of chimney draft is based upon Peclet's hypothesis that the flow through the furnace flues and chimney may be represented by the equation

$$h = \frac{u^2}{64.4}\left(l + G + \frac{fl}{m}\right), \tag{27}$$

in which

h = the head of fluid producing the flow, feet.

u = velocity of the gases in the chimney, feet per second.

G = a coefficient to represent the resistance to the passage of air through the coal.

l = total length of the path of the gases, feet.

m = area of cross section divided by the perimeter.

f = a coefficient depending upon the nature of the surfaces over which the gases pass.

From experiments on chimneys and boilers Peclet gives in connection with this theory the following values of coefficients G and f:

$$G = 12, \qquad f = 0.012,$$

on the basis of 20 to 24 pounds of coal burned per square foot of grate surface per hour. On account of the variation in practice of the factors u, f, and G and the difficulty of determining them engineers prefer to use the modified formulas given further on.

The difference of pressure, or intensity of draft may be expressed theoretically, ignoring friction, as follows:

Let H = height of chimney in feet.
T = absolute temperature of the freezing point, degrees F.
T_1 = absolute temperature of the gases in the chimney.
T_2 = absolute temperature of the outside air.
P = average atmospheric pressure.
P_2 = observed atmospheric pressure.
W = weight of a cubic foot of air at 32 degrees F. and pressure P.
W_1 = weight of a cubic foot of chimney gas at 32 degrees F. and pressure P.

Then the weight of a cubic foot of hot gas in the chimney will be

$$W_1 \frac{P_2}{P} \cdot \frac{T}{T_1} \tag{28}$$

and the weight of a cubic foot of cold air outside will be

$$W \frac{P_2}{P} \cdot \frac{T}{T_2}. \tag{29}$$

The weight of a column of hot gas H feet high and one foot square will be

$$W_1 H \frac{P_2}{P} \cdot \frac{T}{T_1}. \tag{30}$$

Similarly the weight of the cold-air column will be

$$W H \frac{P_2}{P} \cdot \frac{T}{T_2} \tag{31}$$

and the difference in pressure or the intensity of draft will be

$$D = H \frac{P_2}{P} \left(\frac{WT}{T_2} - \frac{W_1 T}{T_1}\right), \tag{32}$$

where D is in pounds per square foot.

By making $P = P_2 = 14.7$, $T = 493$, $W = 0.0807$, $W_1 = 0.084$, and D_1 = pressure in inches of water ($D_1 = 0.192\ D$), equation (32) assumes the familiar form

$$D_1 = H \left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right). \tag{33}$$

By assuming $W = W_1 = 0.081$ and $P = 14.7$, equation (32) may be written

$$D_1 = 0.52\ H P_2 \left(\frac{1}{T_2} - \frac{1}{T_1}\right). \tag{34}$$

This latter form is ordinarily used where the atmospheric pressure differs considerably from that at sea level, as at high altitudes. Table 23 gives the density of air and chimney gases at various temperatures.

Example : Required the maximum theoretical draft obtainable from a chimney 150 feet high, atmospheric pressure 14.7 pounds per square inch, temperature outside air 60 degrees F., temperature chimney gases 550 degrees F.

Here $H = 150$, $T_2 = 461 + 60 = 521$, $T_1 = 461 + 550 = 1011$.

Substituting these values in equation (34),

$$D_1 = 150\left(\frac{7.64}{521} - \frac{7.95}{1011}\right) = 1.02 \text{ inches of water,}$$

which is about 25 per cent greater than the draft actually obtained, and represents the maximum possible under the given conditions, neglecting the resistance offered by the chimney and the pressure

TABLE 23.

DENSITY AND SPECIFIC VOLUME OF AIR AND CHIMNEY GASES AT VARIOUS TEMPERATURES.

Air.				Chimney Gases.					
t	*s*	*v*	*d*	*t*	*d*	*t*	*d*	*t*	*d*
0	11.581	.935	.086353	200	.06334	430	.04695	660	.03730
5	11.706	.945	.085424	210	.06239	440	.04643	670	.03697
10	11.832	.955	.084513	220	.06147	450	.04592	680	.03665
15	11.931	.965	.083623	230	.06058	460	.04542	690	.03633
20	12.085	.976	.082750	240	.05971	470	.04493	700	.03602
25	12.211	.986	.081895	250	.05887	480	.04445	710	.03571
30	12.337	.996	.081058	260	.05805	490	.04398	720	.03540
32	12.387	1.000	.080728	270	.05726	500	.04353	730	.03511
35	12.463	1.006	.080238	280	.05648	510	.04308	740	.03481
40	12.589	1.016	.079434	290	.05573	520	.04264	750	.03453
45	12.715	1.026	.078646	300	.05499	530	.04221	760	.03424
50	12.841	1.037	.077874	310	.05428	540	.04178	770	.03396
55	12.967	1.047	.077117	320	.05358	550	.04137	780	.03369
60	13.093	1.057	.076374	330	.05290	560	.04096	790	.03342
62	13.144	1.061	.076081	340	.05224	570	.04056	800	.03316
65	13.220	1.067	.075645	350	.05159	580	.04017	900	.03072
70	13.346	1.077	.074930	360	.05096	590	.03979	1000	.02861
75	13.472	1.087	.074229	370	.05035	600	.03942	1100	.02678
80	13.598	1.098	.073541	380	.04975	610	.03905	1200	.02516
85	13.724	1.108	.072865	390	.04916	620	.03869	1300	.02373
90	13.851	1.118	.072201	400	.04859	630	.03833	1400	.02245
95	13.976	1.128	.071550	410	.04803	640	.03798	1500	.02131
100	14.102	1.138	.070910	420	.04749	650	.03764	1800	.01848
110	14.354	1.159	.069665	......	....	......	....	2000	.01698

d = density, pounds per cubic foot.

t = temperature, degrees F.

s = specific volume, cubic feet per pound.

v = comparative volume, volume at 32° = 1.

Density of chimney gas taken 0.085 pound per cubic foot at 32° F. and 29.92 inches of mercury.

(Rankine, "Steam Engine," gives the density at 32° F. as varying from 0.084 to 0.087.)

TABLE 24.

THEORETICAL DRAFT PRESSURE IN INCHES OF WATER. CHIMNEY 100 FEET HIGH.[1]

Temp. in the Chimney.	Temperature of the External Air — Barometer, 14.7 Pounds per Square Inch.[2]										
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
200	.453	.419	.384	.353	.321	.292	.263	.234	.209	.182	.157
220	.488	.453	.419	.388	.355	.326	.298	.269	.244	.217	.192
240	.520	.488	.451	.421	.388	.359	.330	.301	.276	.250	.225
260	.555	.528	.484	.453	.420	.392	.363	.334	.309	.282	.257
280	.584	.549	.515	.482	.451	.422	.394	.365	.340	.313	.288
300	.611	.576	.541	.511	.478	.449	.420	.392	.367	.340	.315
320	.637	.603	.568	.538	.505	.476	.447	.419	.394	.367	.342
340	.662	.638	.593	.563	.530	.501	.472	.443	.419	.392	.367
360	.687	.653	.618	.588	.555	.526	.497	.468	.444	.417	.392
380	.710	.676	.641	.611	.578	.549	.520	.492	.467	.440	.415
400	.732	.697	.662	.632	.598	.570	.541	.513	.488	.461	.436
420	.753	.718	.684	.653	.620	.591	.563	.534	.509	.482	.457
440	.774	.739	.705	.674	.641	.612	.584	.555	.530	.503	.478
460	.793	.758	.724	.694	.660	.632	.603	.574	.549	.522	.497
480	.810	.776	.741	.710	.678	.649	.620	.591	.566	.540	.515
500	.829	.791	.760	.730	.697	.669	.639	.610	.586	.559	.534
550	.863	.828	.795	.762	.731	.700	.671	.644	.618	.593	.585
600	.908	.873	.839	.807	.776	.746	.717	.690	.663	.638	.613

1. For any other height multiply the tabular figure by $\frac{H}{100}$, where H is the height in feet.

2. For any other pressure multiply the tabular figure by $\frac{P}{14.7}$, where P is the barometric pressure in pounds per square inch.

required to impart velocity to the gases. Table 24 has been computed from formula (34), and gives the maximum theoretical draft in a chimney 100 feet high for different flue-gas temperatures.

The intensity of draft required to produce best results depends upon the kind and condition of fuel, the thickness of fire, character of grate, and resistance of the breeching, tubes, baffles, dampers, etc. As stated above, the loss of draft in the chimney proper approximates 20 per cent of the total, that in the breeching is taken as 0.1 inch per 100 feet of flue, and 0.05 inch for each right-angle bend; the loss in the boiler varies from 0.3 to 0.6 inch, depending upon the type;* the loss in the furnace varies between wide limits, and depends upon the kind of fuel and the rate of combustion. The curves in Fig. 116 compiled by the Stirling Company and published in their book "Stirling" give the furnace drafts necessary to burn various kinds of fuels at different combustion rates. and give an idea of the influence of the character of the fuel and the rate of combustion.

* Specific figures may be obtained from the manufacturers.

FIG. 116. Relation between Draft and Rates of Combustion.

Example: Determine the probable draft necessary to burn 30 pounds bituminous run of mine per hour per square foot of grate when the outside air is 60 degrees F., the temperature of the chimney gases 550 degrees, and the flue is 100 feet long, with two right-angle bends.

The losses will be divided approximately as follows:

	Inch.
Loss in furnace (from curves in Fig. 116)	0.17
Loss in boiler (average)	0.40
Loss in flue, 100 feet at 0.10 per 100	0.10
Loss in turns, 2 × 0.05	0.10
	0.77

Since the loss in the chimney alone approximates 20 per cent of the total, 0.77 ÷ .80 = 0.96 will be the theoretical draft necessary. From equation (33),

$$D_1 = H\left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right).$$

Substituting for the given values of D_1, T_1, and T_2 in above equation,

$$0.96 = H\left(\frac{7.64}{521} - \frac{7.95}{1011}\right).$$

From which,

$H = 142$, height of stack necessary to produce a draft of 0.17 inch in the furnace.

Table 25 gives the results of a test of a 100-foot unlined steel chimney, showing the variation in draft at different points in the stack.

Theory of Chimney Draft: Power, Oct., 1896, p. 18, Dec., 1898, p. 20, March, 1906, Feb., 1900, p. 12; Engr. U.S., Jan. 15, 1903, May 15, 1902, p. 313; Trans. A.S.M.E., 11–451, 762, 772, 974, 984; Elec. Rev., Lond., Oct. 14, 1904.

128. Chimney Formulas. — Rational methods of determining the height and area of chimneys being cumbersome and unwieldy and of doubtful value for practical use, the various empirical formulas outlined in Table 26 are quite commonly used. They give good results within the limits of the assumptions upon which they are based, but otherwise may lead to absurd results, their applicability depending largely upon the available data covering the various losses with the particular kind, quality, and condition of coal, and conditions of operation. Occasionally practical and local considerations fix the height of the stack irrespective of theoretical deductions. The logical procedure is to *determine first the height of chimney necessary to produce the draft at the desired maximum rate of combustion,* and then to proportion the area by such formulas as (2), (4), or (5), to suit the quantity of fuel to be burned.

The following heights have been found to give good results in plants of moderate size:

	Feet.
With free-burning bituminous coal	80
With anthracite, medium and large sizes	100
With slow-burning bituminous	120
With anthracite pea	130
With anthracite buckwheat	150
With anthracite slack	175

TABLE 25.

CHIMNEY DRAFT.

Test of a 100-Foot Unlined Steel Chimney 3 Feet in Diameter at Massachusetts Institute of Technology. (Peabody & Miller, "Steam Boilers," p. 121.)

	Draft, Inches of Water.		Temperature, Fahrenheit.	
	Maximum.	Minimum.	Maximum.	Minimum.
Over the grate	0.24	0.218	....	....
At the bridge wall	0.382	0.372	....	....
Half-way between bridge and back end of boiler	0.410	0.374	....	....
At the back end of boiler	0.354	0.334	....	....
In uptake near boiler	0.572	0.543	403	389
In stack 34 feet above grate	0.440	0.414	396	374
In stack 51 feet above grate	0.334	0.312	380	368
In stack 68 feet above grate	0.216	0.168	370	354
In stack 85 feet above grate	0.122	0.086	345	314

The chimney serves two 80-horse-power boilers. During test one was banked and the combustion at the grate of the working boiler was 19.8 pounds per square foot of grate surface per hour. Coal burned per hour 590 pounds.

For plants of 800 horse power or more the height of stack should never be less than 150 feet, regardless of the kind of coal used.

Referring to Table 26, formulas (1), (2), (6),(7), and (9) are based upon a fuel consumption of 13 to 15 pounds of anthracite and 22 to 26 pounds of bituminous coal per square foot of grate area per hour. In formulas (3), (4) and (9), the diameter is dependent solely upon the quantity of coal burned per hour and the height is determined mainly by the rate of combustion per square foot of grate. The results accord well with practice. With western coals formula (3) gives results rather too large and the constant should be 120 instead of 180. Formula (5) is

perhaps the most used and has met with much approval. It is based on the assumptions that

1. The draft of the chimney varies as the square root of the height.

2. The retardation of the ascending gases by friction may be considered due to a diminution of the area of the chimney or to a lining of the chimney by a layer of gas which has no velocity and the thickness of which is assumed to be 2 inches. Thus, for square chimneys,

$$E = D^2 - \frac{8\,D}{12} = A - \frac{2}{3}\sqrt{A}, \tag{35}$$

and for round chimneys,

$$E = \frac{\pi}{4}\left(D^2 - \frac{8\,D}{12}\right) = A - 0.591\sqrt{A}. \tag{36}$$

For simplifying calculations the coefficient of $\sqrt{A}$ may be taken as 0.6 for both square and round chimneys, and the formula becomes

$$E = A - 0.6\sqrt{A}. \tag{37}$$

3. The horse-power capacity varies as the effective area E.

4. A chimney should be proportioned so as to be capable of giving sufficient draft to permit the boiler to develop much more than its rated power in case of emergencies or to permit the combustion of 5 pounds of fuel per rated horse power per hour.

5. Since the power of the chimney varies directly as the effective area E and as the square root of the height H, the formula for horse power for a given size of chimney will take the form

$$\text{H.P.} = CE\sqrt{H}, \tag{38}$$

in which C is a constant, found by Mr. Kent to be 3.33, obtained by plotting the results from numerous examples in practice.

The formula then assumes the form

$$\text{H.P.} = 3.33\,E\sqrt{H} \tag{39}$$

or

$$\text{H.P.} = 3.33\,(A - 0.6\sqrt{A})\sqrt{H}, \tag{40}$$

from which

$$H = \left(\frac{0.3\ \text{H.P.}}{E}\right)^2. \tag{41}$$

Table 27 has been computed from equation 5, Table 26.

Many engineers simply adopt the following proportions :

Internal area of chimney at top, one-seventh grate area for bituminous coal.

TABLE 26.—CHIMNEY FORMULAS.

Index.	Author.	References.	Formulas. Horse Power.	Formulas. Height, Feet.	Area, Square Feet.
1	Adams.	Adams, "Handbook for Mechanical Engineers," p. 155.	....	$H=\left(\frac{F}{14\,A}\right)^2.$	$A=\frac{F}{14\sqrt{H}}.$
2	Christie.	Christie, "Chimney Design," p. 22.	$HP=3.24\,A\sqrt{H}.$	$H=\left(\frac{F}{KA}\right)^2.$	$A=\frac{F}{K\sqrt{H}}.$
3	Gale.........	Trans. A.S.M.E., Vol. XI, p. 463.	....	$H=\frac{180}{t}\left(\frac{F}{G}\right)^2.$	$A=0.07\,F^{\frac{4}{5}}.$
4	"Ingenieurs Taschenbuch"	....	....	$H=0.216\left(\frac{F}{G}\right)^2+6\,D.$	....
5	Kent.........	Trans. A.S.M.E., Vol. XII, p. 81.	$HP=3.33\,E\sqrt{H}.$	$H=\frac{\overline{0.3\,HP}^2}{E^2}.$	$E=\frac{0.3\,HP}{\sqrt{H}}.$
6	Lange........	Eng. Record, July 20, 1901, p. 52.	....	$H=15\,D+32.5.$	$A=0.00049\,BF.$
7	Molesworth...	Molesworth, "Pocket Book" ...	$HP=1.28\,A\sqrt{H}.$	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{F}{12\sqrt{H}}.$
8	Nagle........	Power, November, 1902, p. 29...	$HP=2\,D^2\sqrt{H}.$	$H=\frac{\overline{HP}^2}{4\,D^4}.$	$D=\left(\frac{HP}{2\sqrt{H}}\right)^{\frac{1}{2}}.$
9	Nystrum.	Nystrum, "Mechanics," ed. 1882, p. 423.	$HP=1.45\,A\sqrt{H}.$	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{HP}{1.45\sqrt{H}}.$
10	Rankine......	Christie, "Chimney Design," p. 9.	....	$H=\frac{13\,\frac{V}{2\,g}\left(\frac{w}{n\delta}\right)^2\left(\frac{T_1}{T_0}\right)^2}{0.96\,\frac{T_1}{T_2}-1-\frac{0.06\,D}{2\,gA}\left(V^2\,\frac{wT_1}{n\delta T_0}\right)^2}.$	....
11	Smith........	Smith, "Boiler Practice," p. 423.	....	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{0.0825\,F}{\sqrt{H}}.$
12	Stirling......	1905 issue of "Stirling," Stirling Boiler Company.	....	$H=\frac{d'}{0.8\,M}.$	$D'=C\,(HP)^{0.4}.$

H. Height above grate; ft.
A. Inside area at top; sq. ft.
E. Effective area $=A-0.6\sqrt{A}$.
D. Inside diameter at top; ft.
HP. Boiler horse power.
F. Lbs. coal burned per hour.

G. Grate surface; sq. ft.
d'. Net available draft; in. H_2O.
$K=\begin{cases}\frac{F}{1.89\,G} & \text{for bituminous coal.}\\ \frac{F}{G} & \text{for anthracite.}\end{cases}$

$C=\begin{cases}4.68 \text{ for unlined stack.}\\ 4.92 \text{ for lined stack.}\end{cases}$
t. Temperature flue gas; deg. F.
M. Theoretical draft per foot of chimney height.
B. Weight of air per lb. of coal.
V. Cu. ft. air per lb. of coal, 32° F.

w. Lb. fuel burned per second.
n. Ratio grate to chimney area.
δ. Weight of 1 cu. ft. chimney gas, 32° F.
T_2. Temperature external air; deg. F., abs
T_1. Temperature flue gas; deg. F., abs.
T_0. $=461$.
D' = Diameter in inches.

TABLE 27.

SIZE OF CHIMNEYS FOR STEAM BOILERS.

Kent's Formula.

Diam. Inches.	Area (a) Sq. Ft.	Effective Area $E=A-0.6\sqrt{A}$ Sq. Ft.	Height of Chimney.														Equivalent Sq. Chimney Side of Sq. $\sqrt{E}+4$ Inches.
			50 ft.										200 ft.	225 ft.	250 ft.	300 ft.	
			Commercial Horse Power of Boiler.*														
18	1.77	.97	23	25	27	29											16
21	2.41	1.47	35	38	41	44											19
24	3.14	2.08	49	54	58	62	66										22
27	3.98	2.78	65	72	78	83	88										24
30	4.91	3.58	84	92	100	107	113	119									27
33	5.94	4.48		115	125	133	141	149	156								30
36	7.07	5.47		141	152	163	173	182	191	204							32
39	8.30	6.57			183	196	208	219	229	245							35
42	9.62	7.76			216	231	245	258	271	289	316						38
48	12.57	10.44				311	330	348	365	389	[illegible]						43
54	15.90	13.51					427	449	472	503	551	595					48
60	19.64	16.98					536	565	593	632	692	[illegible]					54
66	23.76	20.83						694	728	776	849	918	981				59
72	28.27	25.08						835	876	934	1,023	1,105	1,181	1,253			64
78	33.18	29.73							1,038	1,107	1,212	1,310	1,400	1,485	1,565		70
84	38.48	34.76							1,214	1,294	1,418	1,531	1,637	1,736	1,830	2,005	75
90	44.18	40.19								1,496	1,639	1,770	1,893	2,008	2,116	2,318	80
96	50.27	46.01								1,712	1,876	2,027	2,167	2,298	2,423	2,654	86
102	56.75	52.23								1,944	2,130	2,300	2,459	2,609	2,750	3,012	91
108	63.62	58.83								2,090	2,399	2,592	2,771	2,939	3,098	3,393	96
114	70.88	65.83									2,685	2,900	3,100	3,288	3,466	3,797	101
120	78.54	73.22									2,986	3,226	3,448	3,657	3,855	4,223	107
132	95.03	89.18									3,637	3,929	4,200	4,455	4,696	5,144	117
144	113.10	106.72									4,352	4,701	5,026	5,331	5,618	6,155	128

* Based on a consumption of [illegible] pounds of fuel per boiler horse power. For any other rate multiply the tabular figure by the ratio of 5 to the maximum expected coal consumption per horse power per hour.

Internal area of chimney at top, one-ninth grate area for anthracite coal.

Example: Determine the area and diameter of a stack for a 2000-horse-power plant to operate under the following conditions: Rated load 2000 horse power; maximum overload 40 per cent of rated; flue 150 feet long, with one right-angle bend; average rate of combustion 20 pounds of bituminous coal per square foot of grate surface per hour; atmospheric temperature 60 degrees F.; flue-gas temperature at overload 600 degrees F.; coal burned per boiler horse power, 4 pounds.

With modern types of steam engines or turbines an overload of 40 per cent has little effect on the economy of the prime mover, and the boiler efficiency is but slightly reduced, but an additional allowance of 25 per cent should be made in estimating the overload combustion rate.

The maximum rate of combustion then will be

$$20 + \left(\frac{40 + 25}{100}\right)20 = 33$$

pounds per square foot of grate surface per hour.

The draft required at the point where the flue enters the chimney, considering the various losses, will be found as follows :

	Inch.
Furnace (see curves, Fig. 116)	0.3
Boiler	0.4
Flue, 150 feet at 0.1 inch per 100 feet	0.15
Turns, 1 at 0.05	0.05
	0.9

From formula (33),

$$D_1 = H\left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right).$$

Substituting the following values :

$$T_2 = 60 + 461 = 521\ ; \qquad T_1 = 600 + 461 = 1061$$

$$D_1 = \text{theoretical draft} = \frac{0.9}{0.8} = 1.12 \text{ inch.}$$

$$1.12 = H\left(\frac{7.64}{521} - \frac{7.95}{1061}\right),$$

whence the necessary height of stack is

$$H = 160 \text{ feet (approximately).}$$

Substituting the value of H in Kent's formula, the effective area is found to be

$$E = \frac{0.3 \text{ H.P.}}{\sqrt{H}} = \frac{0.3 \times 2000}{\sqrt{160}} = 47.5 \text{ square feet,}$$

corresponding to an actual diameter of 93 inches.

Chimney Design: Engr. U.S., Jan. 1, 1907, p. 81, Feb. 1, 1907, p. 174; Am. Elecn., March, 1904, p. 123; Eng. Rec., April 18, 1900, July 27, 1901, pp. 52, 82, Sept. 21, 1901, p. 271, March 1, 1902, p. 195, July 19, 1902, p. 60, Nov. 22, 1902, p. 495, May 5, 1906, p. 549; Power, Jan., 1902, p. 12, Nov., 1902, p. 29, Dec. 1, 1903, p. 66, Dec., 1901, p. 570, Dec., 1905, p. 745; Am. Elecn., Dec., 1901, p. 570; Cassier's Mag., July, 1904, p. 341, Feb., 1906, p. 267; Engr. U.S., Oct. 1, 1899, p. 222, Sept. 1, 1904, p. 591, June 15, 1905, p. 403; Eng. Mag., Oct., 1899, p. 124, Trans. A.S.M.E., 11–451; Eng. News, July 20, 1905, p. 64; Engr. U.S., Jan. 1, 1907, p. 91.

129. Height of Chimneys for Boilers using Oil Fuel. — Experimental data relative to chimneys for boilers using oil fuel are rather meager, and discordant, but a study of a number of recent installations seems to indicate that the area need not exceed 50 per cent of that required by the same boiler using bituminous coal. A height 80 to 90 feet above the grate usually affords sufficient draft to force the boilers 50 per cent above rating, but in a number of large installations the chimneys have been designed on the coal-burning basis so as to provide sufficient capacity in case it proves necessary at a future date to revert to the use of coal.

130. Classification of Chimneys. — Chimneys may be grouped into three classes according to the material of construction:

1. Steel.
2. Reënforced concrete.
3. Masonry.

Steel chimneys have many advantages and are finding much favor in large power plants, especially where economy of space warrants the erection of the stack over the boiler, in which case the structural work of the boiler setting answers for both boiler and chimney. Among the advantages are (1) ease and rapidity of construction; (2) less weight for a given internal diameter and height; (3) less surface exposed to the wind; (4) lower cost; (5) smaller space required; (6) slightly higher efficiency if properly calked, for there can be no infiltration of cold air as is likely through the cracks in masonry. The chief disadvantage is the cost of keeping the stack well painted to prevent rust and the corrosive action of the sulphur in the coal.

Steel chimneys may be

1. Guyed.
2. Self-sustained.

131. Guyed Chimneys. — Guyed sheet-iron or steel chimneys or stacks held in position by guy wires are employed in small sizes on account of their relative cheapness. They seldom exceed 52 inches in diameter and 75 feet in height. A heavy foundation is unnecessary, and the stack may be supported by the boiler breeching. The small short stacks are ordinarily riveted in the shop, ready for erection, larger sizes being shipped in sections and riveted at the place of installation. The guy wires are usually fastened to an angle iron or band at about two-thirds the height, and anchored at a distance from the base equal to the height of the band above the ground.

For very tall stacks two sets of guys are used, from four to six wires being fastened to each band, and designed to withstand a wind pressure of 30 pounds per square foot of projected area of the stack. Turn-buckles are employed to equalize tautness. Table 28 gives the thickness of material, with approximate cost and weight, of guyed stacks of different heights and areas.

132. Self-Sustaining Steel Chimneys. — Steel chimneys over 52 inches in diameter are usually self-supporting. They may be built with or without a brick lining, but the lining is preferred, since it prevents radiation and protects the inside from the corrosive action of the flue gases. Since the lining plays no part in the strength of the chimney, it is made only thick enough to support its own weight, and usually of a low-grade fire brick or carefully burned common brick or both. In average practice the fire brick extends 20 or 30 feet above

TABLE 28.

APPROXIMATE WEIGHT AND COST OF GUYED SHEET-STEEL CHIMNEYS.

Height, Feet.	Diameter, Inches.	Thickness of Shell, B.W.G.	Approximate Weight per Foot, Pounds.
40	18	16	13
45	20	16	14
45	22	14, 16	20, 15
50	24	14, 16	22, 16
50	26	14	23.5
55	28	14	25
60	30	12, 14	34, 27
65	32	12, 14	36, 28
70	34	10, 12	48, 39
75	36	10, 12	51, 41

Approximate cost per pound, 3.5 cents to 6.5 cents, including cost of sections riveted and punched, ready for assembling, the higher figure referring to the smaller stacks.

the breeching, the remainder of the lining being of common brick. In chimneys up to 80 inches internal diameter, the upper course is $4\frac{1}{2}$ inches thick, and increases $4\frac{1}{2}$ inches in thickness for each 30 to 40 feet to the bottom. In larger chimneys about 8 inches is the minimum thickness. The lining is generally set in contact with or close to the shell, though a space of 1 to 2 inches is sometimes left between the brickwork and the shell to allow for expansion. This space is occasionally filled with sand.

Self-sustaining stacks may be straight or tapered, and are generally made with a flared or bell-shaped base whose diameter and length are $1\frac{1}{2}$ to 2 times the internal diameter of the stack. The base is riveted to a heavy cast-iron plate bolted to a concrete foundation of sufficient mass to insure stability.

Fig. 117 gives the details of one of the steel chimneys at the power house of the South Side Elevated Railroad, Chicago, Ill.

133. Thickness of Plates. — The sheet is thickest at the bottom, decreasing toward the top of the stack. The proper thickness for any given section may be determined by treating the shaft as a uniformly loaded cantilever, the stresses being expressed by the equation

$$Ph = S\frac{I}{e} = S\frac{\pi}{32}\left(\frac{D_1^4 - D_2^4}{D_1}\right), \tag{42}$$

in which

P = the total wind pressure in pounds.

h = length of the chimney in inches to the center of wind pressure ($h = L/2$ for a cylindrical chimney).

S = safe stress. A low value of 6000 pounds per square inch for single-riveted joints and 8000 for double-riveted joints is recommended, for the reason that a tube of such large diameter with thin walls will hardly fail by rupture according to the formula, but by flattening and bending.

$\frac{I}{e}$ = sectional modulus.

D_1 = external diameter of the shell, inches.

D_2 = internal diameter of the shell, inches.

For chimneys under 7 feet in diameter and 150 feet in height the thickness of plate should not be less than $\frac{3}{16}$ inch, nor less than $\frac{1}{4}$ inch for larger sizes.

13'-9" INSIDE

Fig. 117. Steel Chimney at the Power House of the South Side Elevated Railroad, Chicago.

It is customary to make the courses about 5 feet in height for convenience in erection.

Table 29 gives the dimensions of self-supporting steel stacks as made by the Riter Conley Company of Pittsburg, who use the following empirical formula in determining the thickness of the shell

$$S_1 = \frac{M}{0.8 D_1^2}, \qquad (43)$$

in which

S_1 = stress per lineal inch of section considered,

M = wind moment in inch-pounds, and

D_1 = diameter of the shaft in inches.

Allowing 8000 pounds per square inch as the safe stress for single-riveted joints and 10,000 for double-riveted joints, the required thickness is found by dividing S_1 by 8000 or 10,000.

Example: Determine the thickness of plate at a section 150 feet from the top of a cylindrical steel stack 12 feet in diameter and 200 feet high. Horizontal seams to be double riveted.

The total wind pressure on the section is

$$150 \times 12 \times 25 = 45{,}000 \text{ pounds.}$$

The moment arm is

$\frac{150}{2} \times 12 = 900$ inches.

$D_1 = 144$ inches; $S = 8000$ pounds per square inch.

TABLE 29.

STEEL STACKS. — SIZES OF RITER CONLEY COMPANY, PITTSBURG.

Diameter of Flue.		Total Height.	Total Weight.	How Made.
Ft.	In.	Ft.	Lb.	
5	6	165	67,000	40 ft. of $\frac{3}{16}$ in., 45 ft. of $\frac{1}{4}$ in., 50 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
7	0	160	79,000	30 ft. of $\frac{3}{16}$ in., 50 ft. of $\frac{1}{4}$ in., 50 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
8	6	150	94,000	60 ft. of $\frac{1}{4}$ in., 60 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
10	0	200	150,000	90 ft. of $\frac{1}{4}$ in., 60 ft. of $\frac{5}{16}$ in., 50 ft. of $\frac{3}{8}$ in.
12	0	200	175,000	35 ft. of $\frac{1}{4}$ in., 35 ft. of $\frac{9}{32}$ in., 35 ft. of $\frac{5}{16}$ in., 35 ft. of $\frac{11}{32}$ in., 35 ft. of $\frac{3}{8}$ in., 25 ft. of $\frac{13}{32}$ in.
11	6	225	232,000	40 ft. of $\frac{1}{4}$ in., 40 ft. of $\frac{9}{32}$ in., 40 ft. of $\frac{5}{16}$ in., 40 ft. of $\frac{11}{32}$ in., 40 ft. of $\frac{3}{8}$ in., 25 ft. of $\frac{7}{16}$ in.
12	0	255	256,000	75 ft. of $\frac{1}{4}$ in., 65 ft. of $\frac{5}{16}$ in., 55 ft. of $\frac{3}{8}$ in., 35 ft. of $\frac{7}{16}$ in., 25 ft. of $\frac{1}{2}$ in.

Substituting these values in equation (42),

$$Ph = S\frac{\pi}{32}\left(\frac{D_1^4 - D_2^4}{D_1}\right).$$

$$\frac{45{,}000 \times 1800}{2} = 8000 \times \frac{3.14}{32}\left(\frac{144^4 - D_2^4}{144}\right).$$

$$D_2 = 143.36.$$

Now

$$t = \frac{D_1 - D_2}{2}$$

$$= \frac{144 - 143.36}{2}$$

$$= 0.32 \text{ inch.}$$

The nearest commercial size lies between nine thirty-seconds and five-sixteenths.

The Riter Conley formula gives for this section

$$S_1 = \frac{M}{0.8\, D_1^2} = \frac{45{,}000 \times 900}{0.8 \times 144^2}$$

$$= 2440 \text{ pounds.}$$

$$t = \frac{S_1}{8000} = \frac{2440}{8000} = .305 \text{ inch.}$$

134. Riveting. — The diameter of rivets should always be greater than the thickness of the plate but never less than one-half inch. The pitch should be approximately $2\frac{1}{2}$ times the diameter of the rivet, and always less than 16 times the thickness of the plate. Single-riveted joints are ordinarily used on all sections except the base, where the joint should be double riveted with rivets staggered, although in very large stacks all horizontal seams are double riveted to give greater stiffness to the shaft.

135. Stability of Steel Chimneys. — The wind being ordinarily the only force tending to overturn the stack, and the chimney being rigidly bolted to the foundation, a condition of stability requires that

$$(W_c + W_F)\frac{D}{3} \text{ be equal to or greater than } P\left(\frac{H}{2} + h\right). \qquad (44)$$

in which

W_C = weight of the chimney in pounds.

W_F = weight of the foundation.

P = total wind pressure in pounds.

$D, H,$ and $h,$ in feet, as indicated in the figure.

Expressed graphically: Lay off GP, Fig. 118, equal to the total wind pressure in direction and amount and acting at the center of pressure of the shaft; lay off GW equal to the weight of the stack and foundation; find the resultant GR and produce it to intersect the base line as at R'; if R' falls within the inner third of the base the stack is stable, provided, of course, that the chimney is properly designed and constructed. Therefore the heavier the combined weight of the chimney and its foundation the more stable the structure. (See also paragraph 140.)

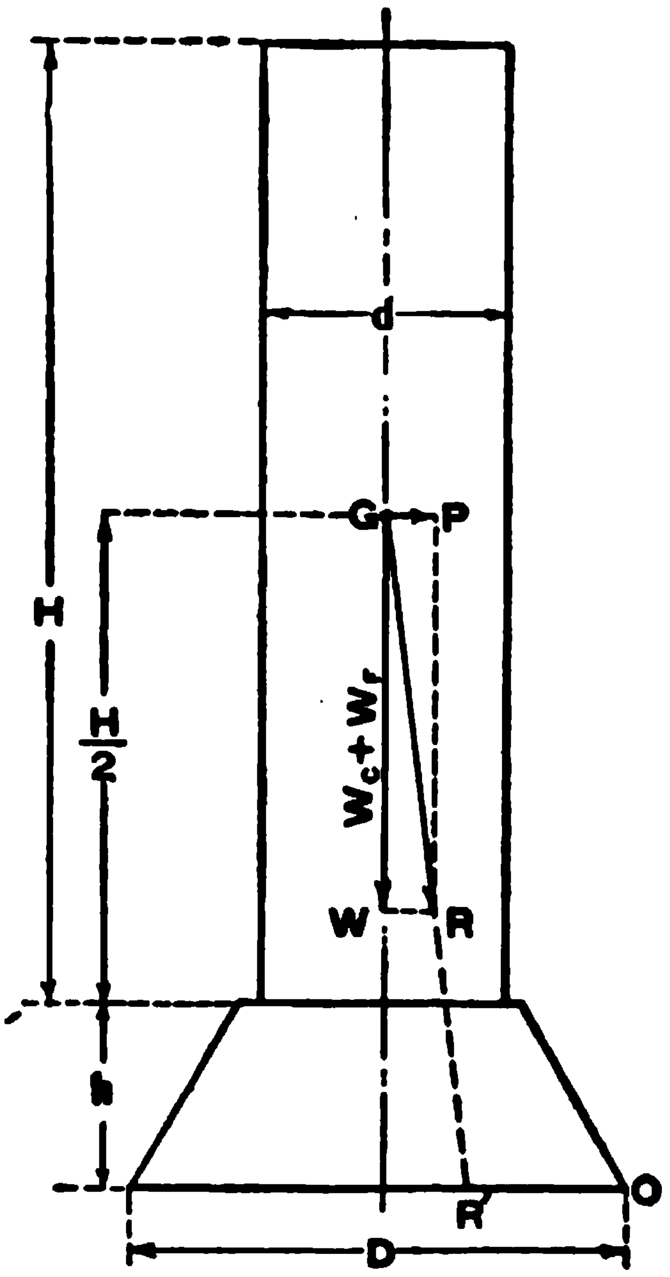


Fig. 118.

D in Fig. 118 varies from one-tenth to one-fifteenth H, depending upon the character of the subsoil. For the ordinary concrete foundation, Christie ("Chimney Design and Theory," p. 57) gives as an average value for D

$$D = \frac{H^2 d}{26,000} + 10. \qquad (46)$$

Detailed Description of Steel Chimneys: Eng. Rec., Feb. 15, 1902, p. 146, July 5, 1902, p. 2, April 23, 1904, p. 523, Sept. 10, 1904, p. 314; Power, Dec., 1905, p. 716, April, 1905, p. 231, Jan., 1902, p. 6; Engr. U.S., Sept. 1, 1904, p. 591, June 15, 1905, p. 403; Eng. News, July 20, 1905, p. 64.

136. Brick Chimneys. — By far the greater number of power-plant chimneys are of brick construction and usually of circular section, though octagonal, hexagonal, and square sections are quite common. The round chimney requires the least weight for stability, and the others in the order mentioned. Taking the total wind pressure on the flat surface of a square stack as unity, the effective pressure for the same projected area will be 0.75 for the hexagonal, 0.6 for the octagonal, and 0.5 for the round.

Brick chimneys may be divided into two general classes:

1. Single shell, Fig. 119, and
2. Double shell, Fig. 120.

The double shell is the most common and consists of an outer shaft of brickwork and an inner core or lining extending part way or throughout the entire length of the shaft.

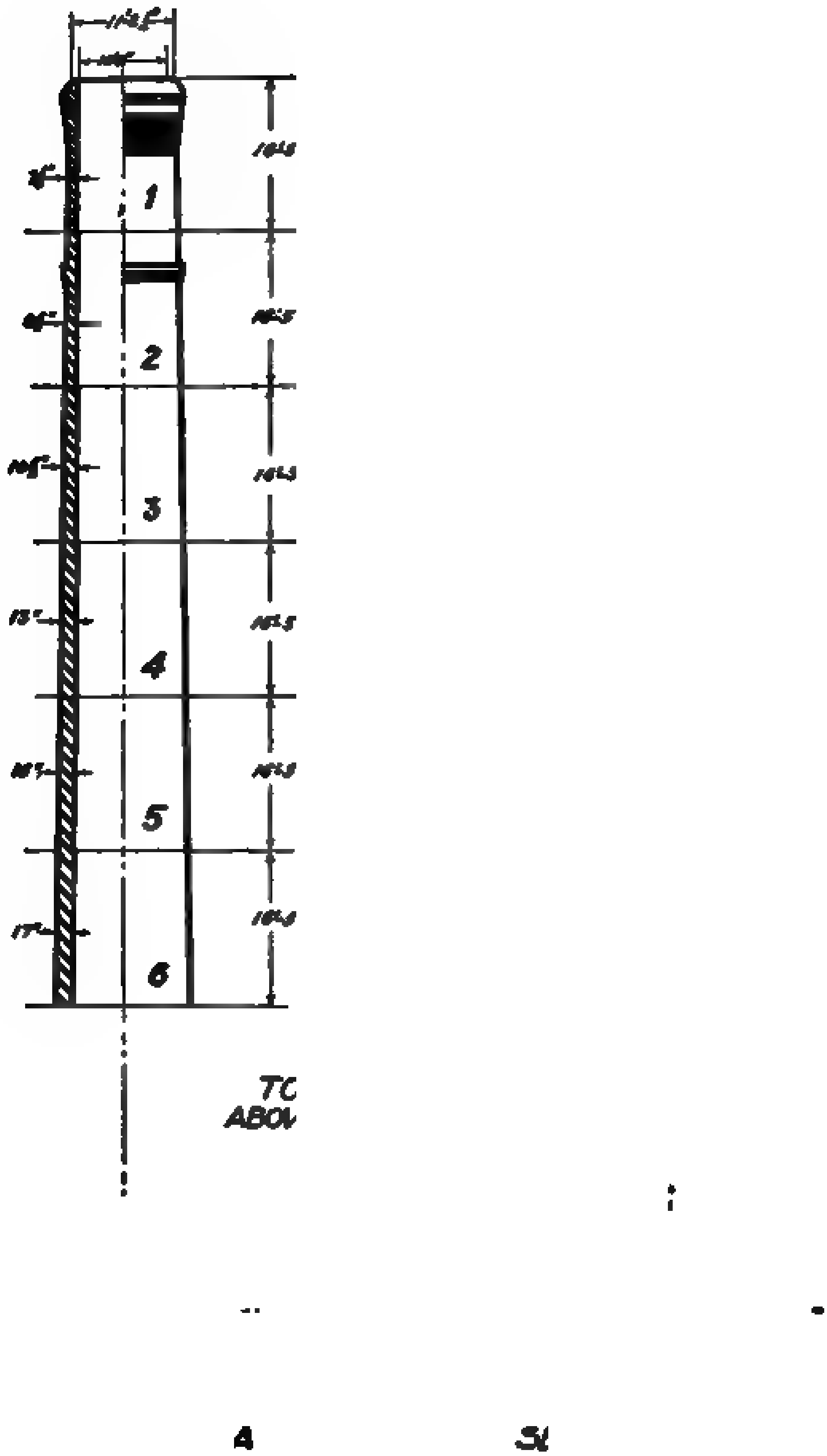


FIG. 119. Custodis Radial Brick Chimney.

The single shell is the general construction where carefully burned and selected brick not easily affected by the heat are used. As the inner core or lining is independent of the outer shell and has no part in the strength of the chimney, the rules for determining the thickness of the walls are practically the same for both single and double shell.

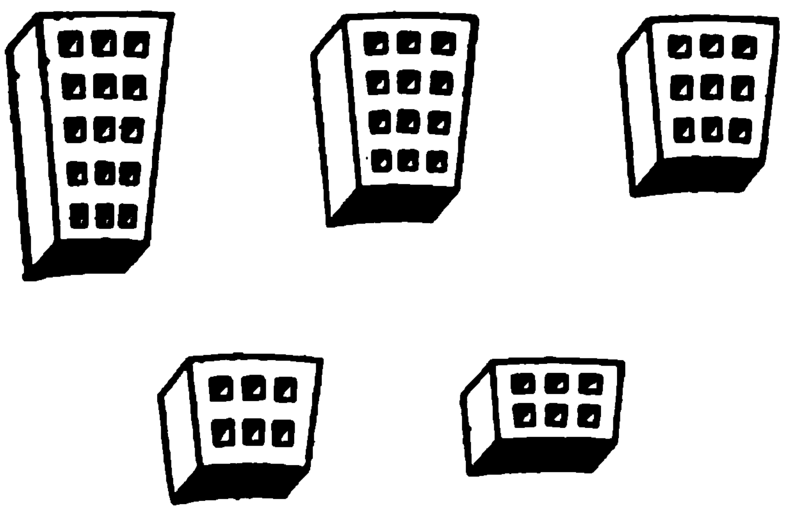


FIG. 119a. Custodis Radial Brick.

137. Thickness of Walls. — The thickness of the wall should be such as to require minimum weight of material for the proper degree of stability, due consideration being paid to the practical requirements of construction. The thickness does not vary uniformly, but decreases from bottom to top by a series of steps or courses as in Fig. 121. In general, the thickness at any section should be such that the resultant stress of wind and weight of shaft will not put the masonry in tension on the windward side or in excessive compression on the leeward side.

For circular chimneys using common red brick for the outer shell the following approximate method gives results in conformity with average practice:

$$t = 4 + 0.05\,d + 0.0005\,H, \tag{47}$$

where

t = thickness in inches of the upper course, neglecting ornamentation, and should, of course, be made equal to the nearest dimension of the brick in use. Ordinary red bricks measure $8\frac{1}{4} \times 4 \times 2$.

d = clear inside diameter at the top, inches.

H = height of stack, inches.

Beginning at the top with this thickness, add one-half brick, or 4 inches, for each 25 or 30 feet from the top downwards, using a batter of 1 in 30 to 1 in 36.

The minimum value of t for stacks built with inside scaffolding should be 7 inches for radial brick and $8\frac{1}{4}$ inches for common brick, as a thinner wall will not support the scaffold. Radial brick for chimneys are made in several sizes, so that the thickness of the walls when they are used increases by about 2 inches at the offsets.

For specially molded radial brick or for circular shells reënforced as in Fig. 120 the length of the different courses may be much less than

FIG. 120. Brick Chimney at the Power Plant of the Armour Institute of Technology.

stated above. The external form of the top is a matter of appearance, and may be designed to suit the taste, but should be protected by a cast-iron or tile cap and provided with lightning rods. Ladders for reaching the top of the chimney are generally located inside of brick stacks and outside of steel ones.

Professor Lang's rule (*Eng. Rec.*, July 20, 1901, p. 53) for determining the length of the different courses is (Fig. 121)

$$h = C\left(20\,t + 60\,i + 0.1056G + 2.5\frac{d}{2} + 656 \tan\alpha - 0.007\,H - 0.453\,p - 18.7\right), \tag{48}$$

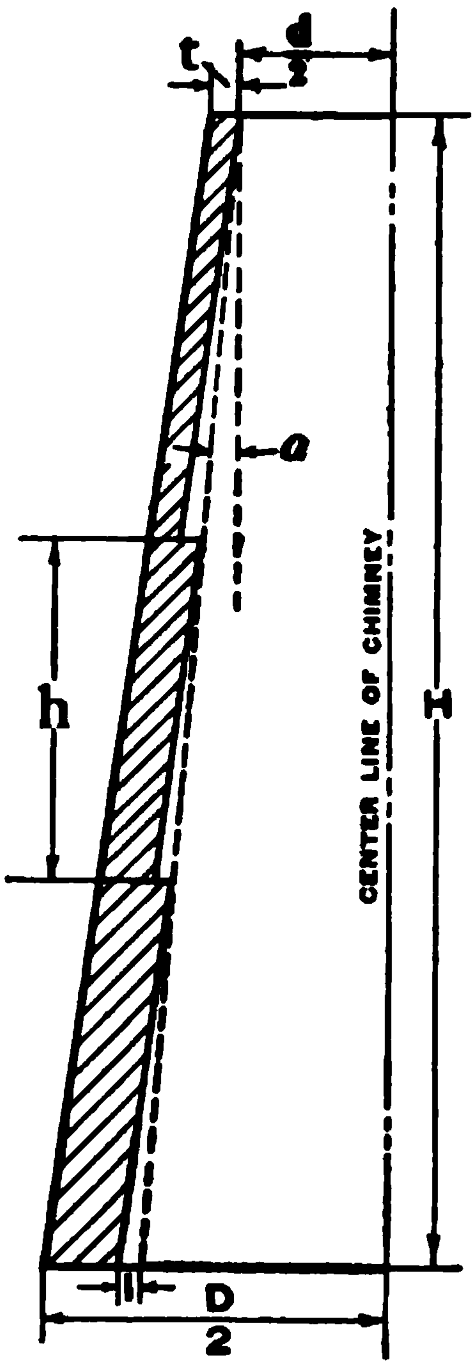


Fig. 121.

in which

h = length of the course under consideration.
C = constant = 1 for a circular, 0.97 for an octagonal, and 0.83 for a square chimney.
i = increase in thickness for each succeeding section in feet.
G = weight per cubic foot of brickwork.
p = wind pressure, pounds per square foot.
α = angle of the internal batter.
All other notations as indicated in Fig. 121.

For chimneys over 100 feet in height he recommends that 100 be used instead of the actual height, since the critical point will be in one of the lower sections and not at the base.

If a value of h is obtained which is not contained an even number of times in H, it may be slightly increased or decreased so as to effect this result.

To determine the stresses at any section the shaft is treated as a cantilever uniformly loaded with a maximum wind pressure of 25 pounds per square foot. If the tension on the windward side subtracted from the compression leaves a positive remainder, the chimney will be stable; if the remainder is negative, the masonry will be in tension, which it withstands but feebly. The sum of the compressive stresses on the leeward side due to wind pressure and weight must be less than the crushing strength of the masonry. The practice, however, of assuming a fixed value for allowable pressure irrespective of the height of the stack gives dimensions that are too low for small

stacks and too high for large stacks. According to Professor Lang, compressive stress on the leeward side in pounds per square inch with single chimneys should not exceed

$$p = 71 + 0.65\,L, \tag{49}$$

where

p = pressure in pounds per square inch.
L = distance in feet from top of chimney to the section in question.

With double shell $p = 85 + 0.65\,L.$ (50)

The tension on the windward side should not exceed,

$$\text{For single shell: } p = (18.5 + 0.056\,L). \tag{51}$$
$$\text{For double shell: } p = (21.3 + 0.056\,L). \tag{52}$$

Example: Determine the maximum stress in the outer fibers of the brickwork at the base of section 8 of the chimney illustrated in Fig. 119 when the wind is blowing 100 miles an hour. Assume the weight of the brickwork 120 pounds per cubic foot.

A wind velocity of 100 miles per hour is estimated to exert a pressure of 50 pounds per square foot on a flat surface and approximately 25 pounds per square foot of projected area on a cylindrical surface. The height of the chimney to section 8 is 131.4 feet. The projected area as computed from the figure is 1800 square feet. Hence p, the total wind pressure, is $1800 \times 25 = 45{,}000$ pounds. The volume of brickwork above section 9 may be calculated, and is 6150 cubic feet, hence the weight $W = 6150 \times 120 = 738{,}000$ pounds.

The area of the joint at this section is 75.3 square feet, therefore the pressure due to the weight of the superimposed brickwork is 738,000 divided by 75.3 = 9800 pounds per square foot. To find the stress due to the wind pressure, substitute the proper values in equation (42):

$$Ph = S\,\frac{I}{e} = 0.0983\left(\frac{D_1^4 - D^4}{D_1}\right)S.$$

Here

P = 45,000 as computed above.
h = 55 feet. (Found by laying out the section and locating the center of gravity.)
D_1 = 16.2.
D = 12.9.

Whence

$$45{,}000 \times 55 = 0.0983\,\frac{16.2^4 - 12.9^4}{16.2}S.$$

From which $S = 9907$ pounds per square foot.

The net stress on any part of the section is the resultant of that due to the weight of the stack and that caused by the wind, the net stress on the windward side being

$$9907 - 9800 = 107 \text{ pounds per square foot,}$$

which is evidently a tensile stress and should never exceed the value given by formula (51):

$$\begin{aligned} p &= (18.5 + 0.056\,L) \\ &= (18.5 + 0.056 \times 131.4) \\ &= 25.8 \text{ pounds per square inch} \\ &= 3715 \text{ pounds per square foot.} \end{aligned}$$

The net compressive stress on the leeward side is 9800 + 9907 = 19,707 pounds per square foot, which should not exceed that given by formula (49):

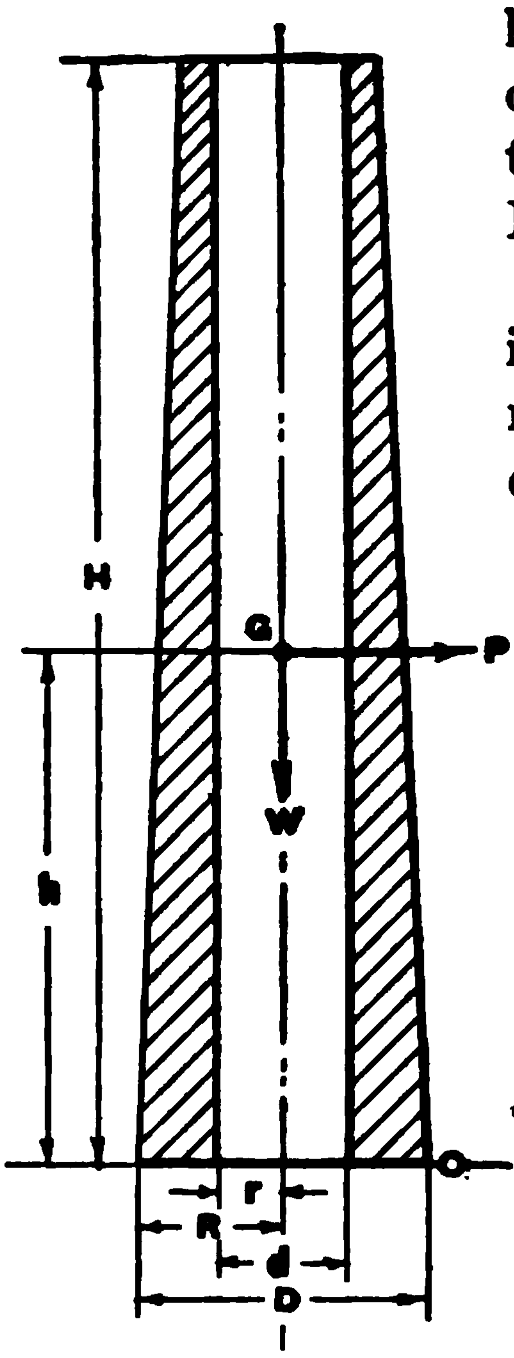
$$\begin{aligned} p &= 71 + 0.65\,L \\ &= 71 + 0.65 \times 131.4 \\ &= 156.4 \text{ pounds per square inch} \\ &= 22{,}521 \text{ pounds per square foot.} \end{aligned}$$

(See also analysis of steel-concrete chimney, paragraph 142.)

138. Core and Lining. — The core or lining of a brick chimney is commonly carried to the top of the shaft, though it sometimes extends only part of the distance. The inside diameter is generally uniform, the offsets being made on the outside. The core and outer shell should be independent to prevent injury due to expansion of the core. The rules for the thickness of lining in steel chimneys apply also to brick chimneys. The batters for the inner and outer shells should be such as to allow at least 2 inches clearance between the two shafts at the top, and the top should be protected by an iron ring or by a projecting ledge from the outer shell.

139. Materials for Brick Chimneys. — Brick for the external shaft should be hard burned, of high specific gravity, and laid with lime mortar strengthened with cement. Lime mortar itself is more resistant to heat, but hardens slowly and may cause distortion in newly erected stacks, and hence should be used only when a long time is taken in building. Mortar of cement and sand alone is not to be recommended, since it does not resist heat well and is attacked by carbon dioxide, particularly in the presence of moisture. A mortar consisting of 1 part by volume of cement, 2 of lime, and 6 of sand

may be used for the upper brickwork, 1, 2½, and 8 respectively for the lower part, and 1, 1, and 4 respectively for the cap. The harder the brick the more cement is necessary, as lime does not cling so well to hard, smooth surfaces. The inner core may be constructed of second-class fire brick, since the temperature seldom exceeds 600 degrees F. Lime mortar is invariably used for the core.



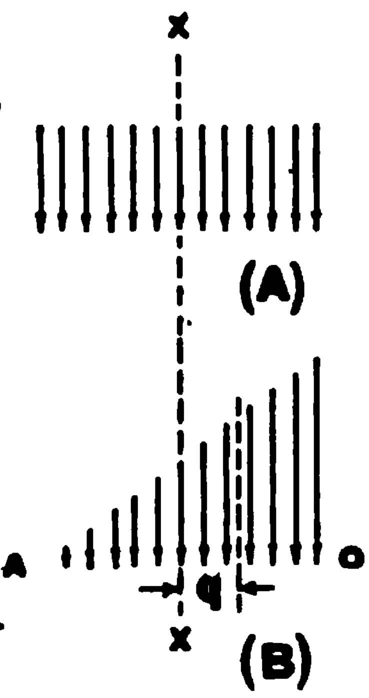


FIG. 122.

140. Stability of Brick Chimneys.—When there is no wind blowing and the chimney is built symmetrically about a vertical axis the pressure due to weight is uniformly distributed over the bearing surfaces, and the center of pressure lies in the line XX, Fig. 122. But when the wind blows the pressure exerted tends to tilt the shaft as a whole-column in the direction of the current, and the pressure decreases from the windward to the leeward side of the base, until, with a sufficiently high velocity of wind, it may become zero, in which case the center of pressure moves a distance q towards the leeward side of the base. As soon as the pressure at A becomes zero the joint begins to open (assuming no adhesion between chimney and base) and the shaft is evidently in the condition of least stability. The distance q through which the center of pressure has moved is called the *radius of the statical moment.* For any column it may be shown that

$$q = \frac{I}{Ae}. \quad \text{(Rankine, “Applied Mechanics,” p. 229.)} \qquad (53)$$

in which

I = moment of inertia of the section.
A = area of the section.
e = distance from the center of the shaft to the outer edge of the joint.

Thus for circular section, $\quad q = \frac{D}{8}.$

For square section, $\quad q = \frac{D}{6}.$

For annular circular ring, $q = \frac{D^2 + d^2}{8 D}$.

For hollow square, $q = \frac{D^2 + d^2}{6 D}$.

The relationship between weight of shaft and wind pressure for the condition of least stability is

$$Ph = Wq, \tag{54}$$

in which

P = total wind pressure, pounds.

h = distance in feet from the base line of the section under consideration to center of gravity of that section.

W = weight of shaft in pounds above the assumed base line.

q = radius of the statical moment.

The condition of least stability for round chimneys requires, therefore, that

$$Ph = W \frac{D^2 + d^2}{8 D} \tag{55}$$

For many purposes it is sufficiently accurate to assume $D = d$, and equation (55) becomes

$$Ph = W \frac{D}{4} \text{ for round chimneys.} \tag{56}$$

$$Ph = W \frac{D}{3} \text{ for square chimneys.} \tag{57}$$

The rule commonly used in Germany, and which is finding much favor with engineers in the United States, gives for the condition of least stability

$$W\left(\frac{1}{2} R + \frac{1}{4} r\right) = Ph. \quad (\textit{Eng. Rec.}, \text{ July 27, 1901, p. 82.}) \tag{58}$$

Notations as in Fig. 122, all dimensions in feet.

This permits of a lighter chimney than equation (55), and the maximum wind pressure may be assumed to put the joint on the windward side in tension or even to permit a slight opening of same.

A rule of thumb for stability is to make the diameter of the base one-tenth of the height for a round chimney; for any other shape to make the diameter of the inscribed circle of the base one-tenth of the height.

The factor of stability is the quotient obtained by dividing the value of q from formula (54) by that from (53). If less than unity, the chimney is in tension at the outer fiber on the windward side, and must be redesigned unless the tension is less than that allowed by equation (51). Calculations for stability should be made for various sections.

Example: Analyze the chimney illustrated in Fig. 119 for stability at say section 8, the following data referring to the portion above the base line of this section.

From the drawing:

Projected area of the stack, 1800 square feet.

Volume of brickwork, 6150 cubic feet.

Outside diameter of base, 16.2 feet.

Inside diameter of base, 12.9 feet.

Center of pressure to base line, 55 feet.

Total height above base line, 131.4 feet.

Maximum total wind pressure:

$$P = 1800 \times 25 = 45{,}000 \text{ pounds.}$$

Weight of shaft:

$$W = 6150 \times 120 = 738{,}000 \text{ pounds.}$$

For stability, according to equation (55),

$$Ph < W\frac{D^2 + d^2}{8\,D}.$$

Substituting the proper values:

$$Ph = 45{,}000 \times 55 = 2{,}475{,}000 \text{ foot-pounds.}$$

$$W\frac{D^2 + d^2}{8\,D} = 738{,}000\left(\frac{\overline{16.2}^2 + \overline{12.9}^2}{8 \times 16.2}\right) = 2{,}441{,}000.$$

While Ph is slightly greater than $W\dfrac{D^2 + d^2}{8\,D}$, for practical purposes the shaft at this section would be called stable under maximum allowable wind pressure.

For stability, according to equation (58),

$$Ph < W\left(\tfrac{1}{2}R + \tfrac{1}{4}r\right).$$

$$Ph = 2{,}475{,}000, \text{ as determined above.}$$

$$W\left(\tfrac{1}{2}R + \tfrac{1}{4}r\right) = 738{,}000\left(\frac{8.1}{2} + \frac{6.45}{4}\right)$$

$$= 4{,}177{,}000.$$

Ph is therefore considerably less than $W(\frac{1}{2}R + \frac{1}{4}r)$, and the condition imposed in equation (58) is more than fulfilled.

Detailed Description of Brick Chimneys: At Paris Exposition, Eng. Rec., Feb. 17, 1900, p. 155; Boston Elevated Ry., Eng. Rec., Dec. 22, 1900, p. 593; Plymouth Cordage Co., Eng. Rec., May 18, 1901, p. 466; Boston, Edison Co., Eng. Rec., Oct. 10, 1903, p. 438; Smelting Works, Freiburg, Germany, Power, Aug., 1900, p. 13; Terminal R.R. Assn. of St. Louis, Mo., Cassier's Mag., Jan., 1898, p. 261; Cambridge Electric Light Co., Engr. U.S., May 15, 1904, p. 331; Interborough Rapid Transit Co., Engr. U.S., Nov. 1, 1904, p. 737; Metropolitan St. Ry. Co., Power, March, 1899, p. 1.

141. Custodis Radial Brick Chimney. — Fig. 119 gives the details of a 200 x 10 foot radial brick chimney constructed of special molded radial brick, formed to suit the circular and radial lines of each section, thus permitting them to be laid with thin, even mortar joints. The blocks are much larger than common brick and the number of joints is proportionately reduced. They are molded with vertical perforations, as shown in Fig. 119a, which permits thorough burning, thereby increasing the density and strength and at the same time reducing the weight of the block. In laying, the mortar is worked into the perforations about one-half inch. The first 60 feet above the base are octagonal in section, with 36-inch walls, and the balance of circular section, with walls tapering gradually from 22 inches to 7⅛ inches in thickness. A radial brick lining extends 60 feet from the base as indicated. The chimney was designed to furnish draft for a 3500-horse-power boiler plant and cost, erected, $8,800. The entire weight of the chimney exclusive of foundation is 870 tons.

Radial brick chimneys without the inner lining are likely to be unduly affected by heat.

The tallest chimney in the world (1907), located at Great Falls, Mont., is of the Custodis type, and is used for leading off the gases from the smelter plant of the Boston and Montana Consolidated Copper and Silver Mining Company. The height above the top of the foundation is 506 feet, and the internal diameter at the top 50 feet. The chimney and foundation cost approximately $200,000.

Custodis Chimney Details: Eng. Rec., Oct. 1, 1904, p. 385; Power, May, 1900, p. 12.

142. Steel-Concrete Chimneys. — The use of concrete reënforced with iron or steel for the construction of chimneys is rapidly increasing. The advantages claimed for this class of stack are:

1. Light weight of the whole structure, being but one-third as great as an equivalent common brick chimney. The space occupied is much less than with either brick or steel stack, on account of the thinness of walls at the base and the absence of any flare or bell.

2. Total absence of joints, the entire structure including foundation being a monolith.

3. Great resisting power against tension and compression.

4. Rapidity of construction. May be erected at an average rate of six feet per day.

5. Adaptability of the material to any form.

This type of chimney being comparatively new, little data concerning depreciation are available, but some which have been in use ten years show little or no deterioration.

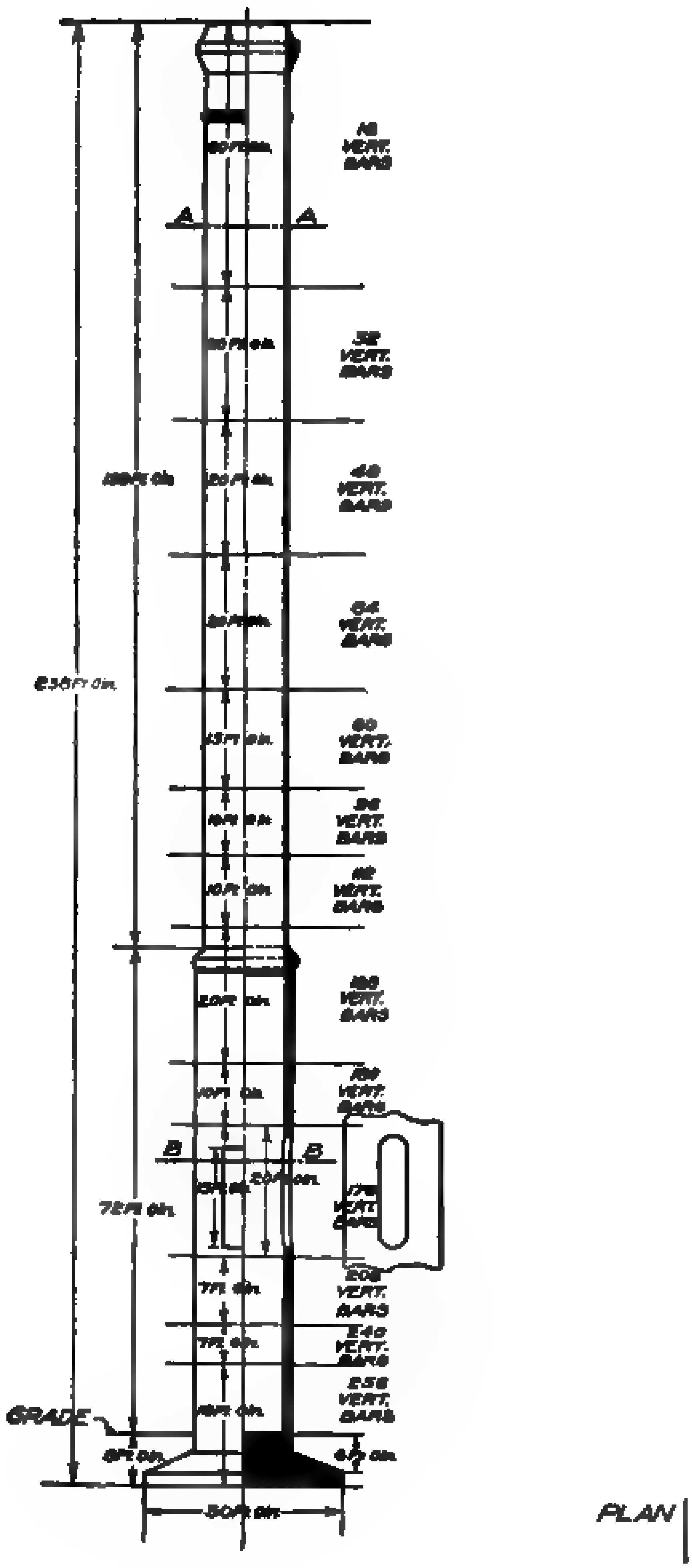


FIG. 123. Weber Reënforced Concrete Chimney.

Fig. 123 gives the details of a Weber steel-concrete chimney erected at Portland, Oregon, for the Portland General Electric Company. The entire structure, foundation and shaft, is a monolith, 238 feet in total height and 12 feet internal diameter, weighing only 889 tons. It occupies but 168 square feet of ground space at the grade level. The weight not including foundation is 470 tons. The stack was erected complete in 58 working days, and cost approximately $13,000.

The cement used was German Portland mixed with select bank sand in proportion of one to three, gravel or crushed stone being used only in the foundation below the ground. The mortar was used medium dry and tamped in the form around the steel reënforcement.

The shaft is of the double-shell type, with inner core extending 70 feet above the grade. The core is but 4 inches in thickness at the base, and the outer shell 8 inches. Both inner and outer shell are reënforced with vertical T bars, $1\frac{1}{4} \times 1\frac{1}{4} \times \frac{3}{16}$ inch, of low-carbon Bessemer steel, spaced at the base 24 inches between centers in the inner core and 4 inches in the outer shell, and increasing in spacing to the top, where the distance between the bars is 12 inches. The horizontal rings are $1 \times 1 \times \frac{1}{8}$ T's spaced 18 inches between centers in the core and 36 inches in the outer shell. The steel bars vary from 16 to 30 feet in length, and where they meet lengthwise are lapped not less than 24 inches. The use of different lengths of steel prevents the laps from concentrating in any given section.

The tallest chimney of this type (1907) was erected for the Butte Reduction Works at Butte, Mont. Its height is 350 feet and inside diameter 18 feet.

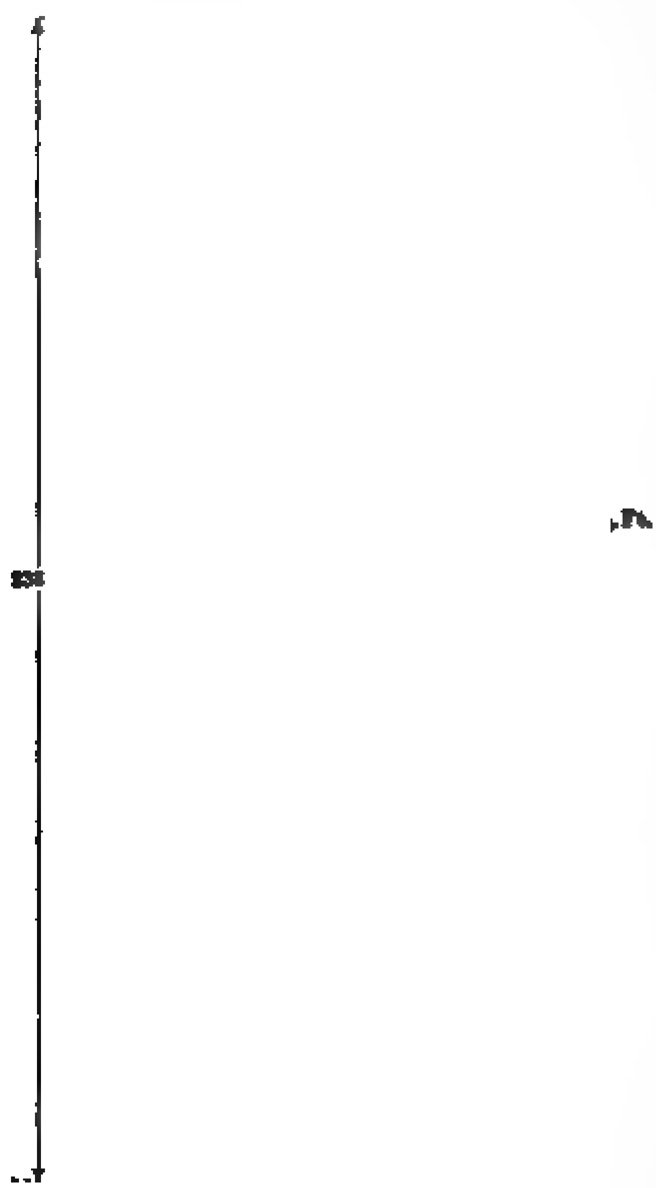


Fig. 124.

The following strain sheet gives the Weber Company's analysis of the chimney illustrated in Fig. 123, and is based on a wind pressure of 50 pounds per square foot. Notations as in Fig. 124.

WEIGHTS.

$$
\begin{aligned}
W_f &= \text{weight of foundation} \\
&= \left(l_1^2 h_1 + \frac{l_1^2 + l_2^2}{2}\, h_2\right) 150 \\
&= \left(30^2 \cdot 2 + \frac{30^2 + 15^2}{2}\, 3\right) 150 \\
&= 523{,}200 \text{ pounds.} \\
150 &= \text{weight per cubic foot of concrete.} \\
W_e &= \text{earth weight on foundation} \\
&= \{l_1^2 h_6 - \text{(volume of foundation)}\}\, 100 \\
&= (7200 \text{ cubic feet} - 3995 \text{ cubic feet})\, 100 \\
&= 320{,}500 \text{ pounds.} \\
100 &= \text{weight per cubic foot of earth.} \\
W &= \text{weight of shaft} \\
&= \{A_1(h_4 + h_3) + A_2\,(h_4 + h_3) + A_3 h_5\}\, 150 \\
&= \{38.5\,(72 + 3) + 13\,(72 + 3) + 19.75 \cdot 158\}\, 150 \\
&= 934{,}950 \text{ pounds.} \\
W_t &= \text{total weight} \\
&= W_f + W_e + W = 1{,}778{,}650 \text{ pounds (889 tons).}
\end{aligned}
$$

SECTION AT GRADE G_r.

I. $$
\begin{aligned}
W_i &= \text{weight of outer shell and single shell above section} \\
&= (A_1 h_4 + A_3 h_5)\, 150 \\
&= (28.5 \cdot 72 + 19.75 \cdot 158)\, 150 \\
&= 775{,}806 \text{ pounds.}
\end{aligned}
$$

II. $$
\begin{aligned}
r &= \text{radius of statical moment} \\
&= \frac{D_1}{8}\left[1 + \left(\frac{d_1}{D_1}\right)^2\right] \\
&= \frac{14.66}{8}\left[1 + \left(\frac{13.33}{14.66}\right)^2\right] \\
&= 3.35 \text{ feet.}
\end{aligned}
$$

III. $$
\begin{aligned}
P &= \text{wind pressure on chimney} \\
&= D_1 h_4 \frac{50}{2} + D_2 h_5 \frac{50}{2} \\
&= 14.66 \times 72 \times 25 + 13 \times 158 \times 25 \\
&= 77{,}738 \text{ pounds.}
\end{aligned}
$$

M = wind moment on section

$$= D_1h_4\frac{50}{2}\times\frac{h_4}{2} + \left(D_2h_5\frac{50}{2}\right)\left(h_4 + \frac{h_5}{2}\right)$$

$$= 14.66 \times 72 \times \frac{50}{2} \times \frac{72}{2} + \left(13 \times 158 \times \frac{50}{2}\right)\left(72 + \frac{158}{2}\right)$$

$= 8,703,818$ foot-pounds.

IV. N = statical moment

$= rW_t.$

$= 3.35 \times 775,806$

$= 2,598,950$ foot-pounds.

V. B = bending moment

$= M - N$

$= 8,703,818 - 2,598,950$

$= 6,104,868$ foot-pounds.

VI. $\frac{I}{e}$ = section modulus

$$= 0.0982\left(\frac{D_1^4 - d_1^4}{D_1}\right)$$

$$= 0.0982\left(\frac{\overline{14.66 \times 12}^4 - \overline{13.33 \times 12}^4}{14.66}\right)$$

$= 169,703.$

VII. z = tension per square inch sectional area

$$= 12B \div \frac{I}{e}$$

$= 12 \times 6,104,868 \div 169,703$

$= 432.5$ pounds.

VIII. Z = total tension

$= 144\ A_1z$

$= 144 \times 28.5 \times 432.5$

$= 1,825,015$ pounds.

IX. s = area steel required

$= \frac{Z}{a}$. (a = sectional strain on steel)

$= 16,000$ pounds per square inch

$= 114.2$ square inches.

X. K = number of bars

$= \frac{s}{x}$. (x = 0.45 square inches = area of one bar)

= 252 bars.

FOR STABILITY.

XI. L = length of one side of base.

$$L = \frac{M}{W_t} \times 6$$

$$= \frac{8{,}703{,}818}{1{,}778{,}650} \times 6$$

= 29.4 feet.

SECTION 42′ 0″ ABOVE GRADE.

I. W_1 = 596,250 pounds.
II. r = 3.35 feet.
III. P = 62,295 pounds. M = 5,761,325 foot-pounds.
IV. N = 1,997,438 foot-pounds.
V. B = 3,763,889 foot-pounds.
VI. $\frac{I}{e}$ = 169,703.
VII. z = 222 pounds.
VIII. Z = 911,088 pounds.
IX. s = 57 square inches.
X. K = 127 bars.

SECTION AT OFFSET.

I. W_1 = 468,000 pounds.
II. r = 3 feet.
III. P = 51,350 pounds. M = 4,056,650 foot-pounds.
IV. N = 1,404,000 foot-pounds.
V. B = 2,652,650 foot-pounds.
VI. $\frac{I}{e}$ = 102,041.
VII. z = 311 pounds.
VIII. Z = 786,000 pounds.
IX. s = 55.5 square inches.
X. K = 123 bars.

SECTION 50′ 0″ FROM TOP.

I. W_1 = 148,125 pounds.

II. r = 3 feet.

III. P = 16,250 pounds. M = 406,250 foot-pounds.

IV. N = 444,365 foot-pounds.

Since the statical moment N is greater than the wind moment M, there is no bending moment B, so no steel is required, the chimney above this section standing of its own weight. However, thirty-two bars are continued to the top.

Detailed Description of Concrete-Steel Chimneys: Weber Chimney, Power, July, 1907, p. 476; Burt Portland Cement Co., Eng. News, Dec. 29, 1904, p. 579; Leiter Coal Mines, Eng. Rec., March 21, 1904, p. 661, Power, Dec., 1904, p. 787; Pacific Electric Ry. Co., Los Angeles, Cal., Eng. News, April 2, 1903, p. 308; Laclede Fire Brick Co., Eng. News, April 2, 1903, p. 310; Tall Concrete Chimneys, Eng. News, July 20, 1905, p. 57, Aug. 3, 1905, p. 120, Feb. 15, 1906, p. 165, Oct. 11, 1906, p. 387; Butte Reduction Works, Eng. Rec., Feb. 3, 1906, p. 124; New Types, Eng. Rec., Dec. 15, 1906, p. 670.

143. Breeching. — The area of the flue or breeching leading from the boilers to the chimney is generally made equal to or a little larger than the internal area of the chimney, 20 per cent greater being an average figure. The flue may be carried over the boilers or back of the setting, or even under the fire-room floor, but in any case should be as short as possible and free from abrupt turns. Short right-angled turns reduce the draft approximately 0.05 inch for each turn, and a convenient rule is to allow 0.1 inch loss for each 100 feet of flue if of circular cross section and constructed of steel, and double this amount for brick flues of square section. The cross section of the flue need not be the same throughout its entire length, but may be tapered and proportioned to the number of boilers. Where two flues enter the stack on opposite sides, a diaphragm is inserted as indicated in Fig. 119. Flues should be covered with heat-insulating material.

144. Chimney Foundations. — On account of the concentration of weight on a small area the foundation of a chimney should be carefully designed. In most cities the building laws limit the maximum loads allowed for various soils and materials, and although they vary considerably the average is approximately as follows:

MATERIAL.	SAFE LOAD, LB. PER SQ. FT.
Hard-burned brick masonry, cement mortar, 1 to 2	20,000–30,000
Hard-burned brick masonry, cement mortar, 1 to 4	18,000–24,000
Hard-burned brick masonry, lime mortar	10,000–16,000
Concrete, 1 to 8	8,000–10,000

Kind of Soil.	Safe Load, Tons per Sq. Ft.
Quicksands and marshy soils	0.5
Soft wet clay	1.0
Clay and sand 15 feet or more in thickness	1.5
Pure clay 15 feet or more in thickness	2.0
Pure dry sand 15 feet or more in thickness	2.0
Firm dry loam or clay	3.0– 4.0
Gravel well packed and confined	6.0– 8.0
Rock broken but well compacted	10.0–15.0
Solid bed rock	Up to ⅛ of its ultimate crushing strength.
	Tons per Pile.
Piles in made ground	2.0
Piles driven to rock or hardpan	25.0

Chimney foundations as a rule are constructed of concrete except where the low sustaining nature of the soil necessitates the use of piles or a grillage of timber or steel. For masonry chimneys the foundation is designed to give the necessary support to the shaft without particular reference to its mass or distribution, as the shape of the foundation has virtually no effect on its stability as a column. In steel and reënforced concrete chimneys the shape and weight of the foundation are a function of the desired factor of stability, since the shaft is securely anchored to the foundation and the two form practically one mass. The foundation should be designed to fulfill the conditions in formula (46) in addition to the requirements for mere support.

Table 30 gives the least diameter and depth of foundation for steel chimneys of various diameters and heights.

145. Chimney Efficiencies. — The chimney as a *mover of air* has a very low thermodynamic efficiency. Compared with that of a fan its performance is very poor, and mechanical-draft concerns sometimes use this as an argument.

Example: A chimney 200 feet high and 10 feet in diameter furnishes draft for a battery of boilers rated at 3500 horse power. Average outside temperature 60 degrees F.; temperature of flue gases 500 degrees F.; calorific value of the fuel 14,000 B.T.U. per pound. Compare the thermal efficiency of the chimney as a mover of air with that of a forced draft apparatus of equivalent capacity.

From Table 24 we find that a chimney 200 feet high, with temperatures as stated above, will furnish a theoretical draft of 1.27 inches, equivalent to a pressure of 6.6 pounds per square foot. Neglecting friction the height H of a column of external air which would produce this pressure is

$$H = \left(\frac{d_1 - d}{d_1}\right) h, \tag{59}$$

in which

h = height of the chimney in feet.
d = density of the hot gases in the stack.
d_1 = density of the outside air.

Substitute in (59),

$$d_1 = 0.0763, \quad d = 0.0435, \quad \text{and } h = 200.$$

TABLE 30.

SIZES OF FOUNDATION FOR STEEL CHIMNEYS.

Diameter, Feet.	Height, Feet.	Least Diameter of Foundation.	Least Depth of Foundation.
3	100	15′ 9″	6′ 0″
4	100	16′ 4″	6′ 0″
4	125	18′ 5″	7′ 0″
5	150	20′ 4″	9′ 0″
5	200	23′ 8″	10′ 0″
6	150	21′ 10″	8′ 0″
6	200	25′ 0″	10′ 0″
7	150	22′ 7″	9′ 0″
7	250	29′ 8″	12′ 0″
9	150	23′ 8″	10′ 0″
9	275	33′ 6″	12′ 0″
11	250	24′ 8″	10′ 0″
11	350	36′ 0″	14′ 0″

$$H = \left(\frac{0.0763 - 0.0435}{0.0763}\right) 200$$
$$= 85.9 \text{ feet.}$$

The theoretical velocity of the air entering the base of the chimney under this head is

$$v = \sqrt{2\,gh}$$
$$= \sqrt{2 \times 32.2 \times 85.9}$$
$$= 74.5 \text{ feet per second.}$$

The weight of the gas escaping per second

$$= 74.5 \times \text{area of the stack} \times 0.0763$$
$$= 446 \text{ pounds.}$$

The displacement of this volume of gas is the result of heating it from 60 to 500 degrees F. Taking the specific heat of the gas as 0.2375, the heat necessary to displace 456 pounds per second is

$$\text{Heat required} = 446 \times 0.2375 \times (500 - 60)$$
$$= 46{,}500 \text{ B.T.U. per second}$$

The work actually performed is that of overcoming a total resistance of $6.6 \times 78.5 = 518$ pounds (78.5 = internal area of the chimney) through a space of 74.5 feet; i.e.,

$$\text{Work done} = 74.5 \times 518 = 38{,}591 \text{ foot-pounds per second,}$$
$$= 49.7 \text{ B.T.U. per second.}$$
$$\text{Efficiency} = \frac{49.7}{46{,}500} = .00107, \text{ or about } \tfrac{1}{10} \text{ of 1 per cent.}$$

If a fan be substituted for the chimney and we allow say 8 per cent for the efficiency of engine and boiler, 40 per cent for the fan, and 25 per cent for friction, the combined efficiency will be

$$0.08 \times 0.40 \times 0.75 = 0.024, \text{ or 2.4 per cent.}$$

The fan then will be $\frac{0.024}{0.00107} = 22.4$ times more efficient than the chimney *as a mover of air.*

146. Cost of Chimneys. — Christie ("Chimney Design and Theory") gives the following costs of chimneys 150 feet high and 8 feet internal diameter:

Common red brick	approximate	cost	$8,500.00
Radial brick	do	do	6,800.00
Steel, self-supporting, full lined	do	do	8,300.00
Steel, self-supporting, half lined	do	do	7,800.00
Steel, self-supporting, unlined	do	do	5,820.00
Steel, guyed	do	do	4,000.00

The following approximate costs of various sizes of a well-known radial brick chimney give an idea of the variation in cost due to increase in diameter and height:

Size of Chimney.		Cost.	Size of Chimney.		Cost.
Height.	Diameter.		Height.	Diameter.	
Feet.	Feet.		Feet.	Feet.	
75	4	$1,350.00	175	8	$7,050.00
75	6	1,950.00	175	10	7,925.00
75	8	2,650.00	175	12	8,950.00
75	10	3,725.00	175	14	9,725.00
125	6	3,500.00	200	8	9,250.00
125	8	4,250.00	200	10	10,500.00
125	10	4,675.00	200	12	11,100.00
125	12	5,125.00	200	14	12,500.00
150	8	6,150.00	250	10	16,500.00
150	10	7,125.00	250	12	18,250.00
150	12	7,750.00	250	14	21,500.00
150	14	8,275.00	250	16	24,250.00

TABLE 31.

PROPORTIONS OF CHIMNEYS FOR FACTORY STEAM BOILERS, COLLECTED FROM PRACTICE. (Hutton.)

Height of Chimney above the Ground in Feet.	Internal Dimensions.		Ratio of Bottom to Top. Internal Area.	Thickness of Walls.	
	Size of Base at the Ground Line.	Size of Top.		Thickness at Base in Inches at Ground Line.	Thickness at the Top in Inches.
40	2′ 6″	1′ 9″ sq.	2.04	18	9
60	2′ 11″	2′ 0″ sq.	2.12	18	9
70	3′ 4″	2′ 3″ sq.	2.13	23	9
80	3′ 8″	2′ 6″ sq.	2.18	28	9
90	4′ 0″	2′ 9″ sq.	2.27	28	9
100	4′ 8″	3′ 0″ diam.	2.40	28	9
110	4′ 10″	3′ 3″ diam.	2.33	28	9
120	5′ 6″	3′ 6″ diam.	2.40	28	9
135	6′ 0″	4′ 0″ diam.	2.30	28	9
150	4′ 6″	3′ 0″ diam.	2.25	28	14
155	6′ 0″	4′ 6″ diam.	1.78	56	14
160	9′ 0″	5′ 0″ sq.	3.24	36	14
170	7′ 6″	5′ 0″ diam.	2.25	36	14
180	6′ 4″	4′ 6″ diam.	2.00	54	14
200	5′ 3″	3′ 6″ diam.	2.28	36	14
225	16′ 0″	6′ 6″ sq.	4.00	36	14
250	19′ 0″	13′ 0″ diam.	2.13	40	14
300	14′ 0″	9′ 0″ diam.	2.42	48	14
450	21′ 6″	10′ 2″ diam.	4.35	59	14

CHAPTER VIII.

MECHANICAL DRAFT.

147. General. — The intensity of natural draft in a chimney depends mainly upon the height of the stack and the temperature of the chimney gases, and the chimney should be designed to meet the maximum requirements, permitting the damper to be partly shut at times. There is usually no practicable means of increasing the draft after the maximum has been reached. Again, chimney draft is peculiarly susceptible to atmospheric influence and may be seriously impaired by adverse winds and air currents. Notwithstanding these apparent limitations, by far the greater number of steam power plants depend upon chimneys for draft, and for obvious reasons as will be discussed later. In many cases artificial draft has a great advantage and under certain conditions is indispensable; it is very flexible and readily adjusted to effect various rates of combustion, irrespective of climatic influences, and permits any degree of overload without undue expenditure of energy.

Artificial draft may be broadly classified under two heads

1. The vacuum or induced draft and
2. The plenum or forced-draft method.

In the former a partial vacuum is produced above the fire by suitable apparatus, and the effect is substantially that of natural draft.

In the forced-draft system pressure is produced in the ash pit, the air being forced through the grate.

In both systems the artificial draft is usually produced by either

1. Steam jets or
2. Centrifugal fans or exhausters.

148. Steam Jets. — Fig. 125 shows an application of a ring jet to the base of a stack. The apparatus is very simple, inexpensive, and easily applied. It consists essentially of a ring or a series of concentric rings of 1-inch or $1\frac{1}{2}$-inch pipe, perforated on the upper side with $\frac{1}{16}$ or $\frac{1}{8}$ inch holes, and placed in the base of the stack, so that the jets are discharged upward, thus creating a draft independent of the temperature of the flue gases. The steam connection to the jet is generally made direct to the boiler and not to the steam main, though the jet is often produced by exhaust steam.

Fig. 126 illustrates a Bloomsburg jet, which involves to some extent the principle of the ejector.

The increase in draft produced by these devices as ordinarily installed

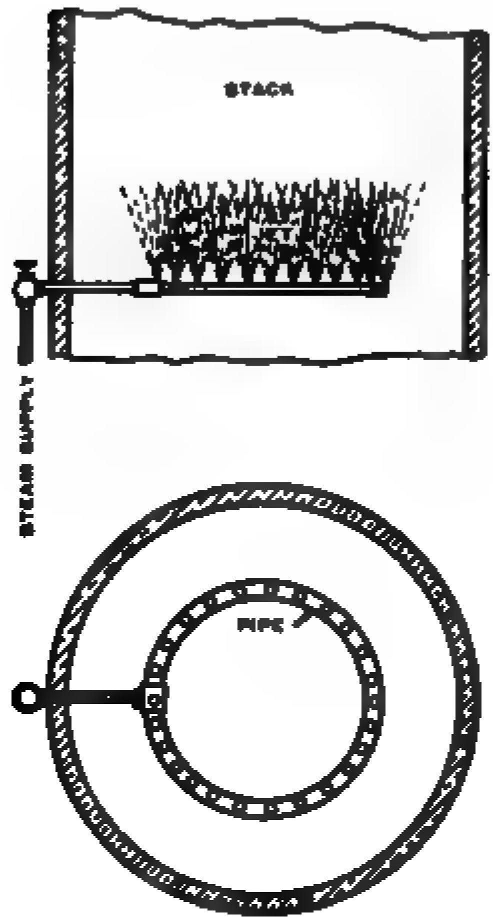


Fig. 125. Ring Steam Jet.

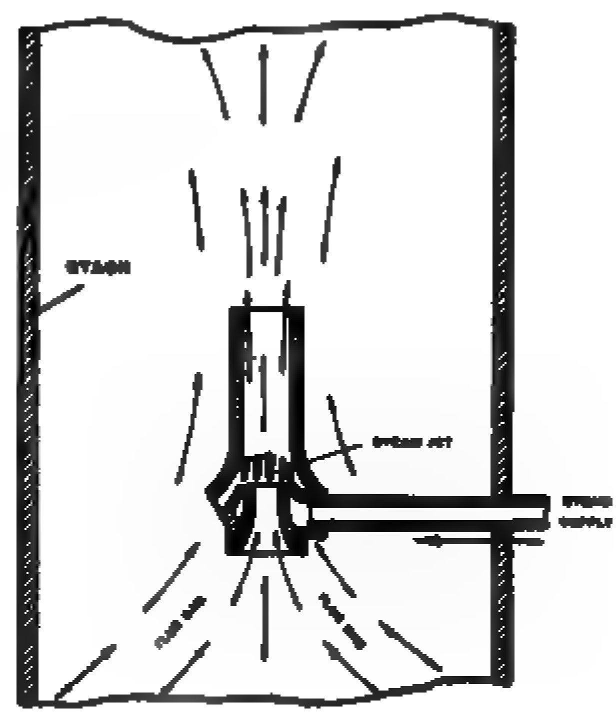


Fig. 126. Bloomsburg Jet.

is not great, although in locomotive practice where the entire exhaust is discharged up the stack an intense draft is obtained.

Fig. 127 shows the application of a "McClaves argand blower."

Fig. 127. McClaves Argand Blower.

The steam is discharged below the grate through a perforated hollow ring, as indicated, drawing the air through the funnel by inspiration. This creates a powerful draft by forming an air pressure in the ash pit,

and is an especially useful system of forcing fires for boilers which need forcing for short periods only.

Steam jets are very uneconomical, since a large amount of steam is required to produce good results. Table 32, based on experiments at the New York Navy Yard to determine the best form of steam jet for producing draft in launch boilers, shows steam consumptions of 8.3 to 21.2 per cent of the total steam made. Table 33 gives the steam consumption of a number of types of steam jet blowers as determined by A. J. Whitham. The best performance is 4.6 per cent and the poorest 11.1 per cent of the total boiler steam generated. Steam jets below the grate are said to prevent clinker from forming where fine anthracite coals are used, and thus to assist in keeping the fire free and open.

Steam jets arranged above the grate and discharging either from the side walls, front wall, or bridge wall, oftentimes assist complete combustion by stirring up the volatile gases and air and insuring a thorough mixture, thus affording one of the simplest and frequently a very efficient means of furthering smokeless combustion. The action is of course purely mechanical, the steam in itself not being a supporter of combustion; hence if the air supply is deficient the steam jet is of no avail unless arranged to carry sufficient air along with the steam.

Fig. 128 illustrates such an application to a hollow bridge wall. The top of the wall is fitted with a small cast-iron column *M*, partially

FIG. 128. Application of Steam Jets to Hollow Bridge Wall.

imbedded in the brickwork. A series of 1-inch holes "*OO*," drilled near the top of the casting, furnish exits for the steam and air. A steam jet in one end of the column induces air into the iron chamber and forces it across the fire in fine streams. Excessive air dilution is avoided by partially closing the ash-pit doors and by regulating the intensity of the jets. An installation of this type is especially effective in connection with coal having a tendency to fuse and seal the air pas-

sages in the grate. Two Stirling boilers at the Armour Institute of Technology equipped with this device gave practically smokeless combustion at all normal loads, though at heavy overloads it was sometimes necessary to slightly open the fire doors. Without the use of the jets smoke could not be prevented even at light loads. Analysis of the flue gases showed but a slight decrease in the percentage of CO_2.

TABLE 32.

RESULTS OF EXPERIMENTS UPON STEAM JETS AT NEW YORK NAVY YARD.*

Index of Jet.	Pounds of Water Evaporated per Hour.				
	A	B	C	D	E
In boiler making steam	463.8	580.0	361.25	528.5	545.00
In boiler supplying jets	97.5	120	30	63.2	76.25
Per cent of steam used by jet	21.2	20.7	8.3	12.0	19.0

* Annual Report of the Chief of the Bureau of Steam Engineering, U. S. Navy, 1890.

TABLE 33.

CONSUMPTION OF STEAM BLASTS COMPARED.†

Coal.	Name of Blower.	Per Cent of Air Openings in Grate.	Pounds of Dry Coal burned per Hour per Square Foot of Grate.	Per Cent of Total Steam Generated in the Boilers that is required to operate the Steam Blasts.
Rice	Young	11	25.8	11.1
Do	...do	11	17.9	7.0
Do	Wilkinson	7	27.0	10.8
Buckwheat	Young	11	27.3	10.8
Do	....do	11	16.7	4.6
Do	....do	26	31.4	8.9
Do	McClave	11	16.4	6.7
Do	....do	11	26.1	9.3
Do	Wilkinson	7	32.5	7.8
Do	....do	7	45.4	10.2

† Trans. A.S.M.E., Vol. XVII. — Whitham.

149. Parson Smokeless Furnace. — The Parson forced-draft system for smokeless combustion, applied to a return tubular boiler as illustrated in Fig. 129, comprises a specially designed grate *G*, depending upon a steam jet blower *A* for draft. Part of the steam is admitted

below the grate and part over the fire through the hollow bridge wall *H*. The supply of air above the grate is regulated by means of damper *F*. The steam to blower *A* is automatically adjusted by regulator *N*, which is actuated by the steam pressure. The steam to the jet is superheated by passing the supply pipe through the setting as indi-

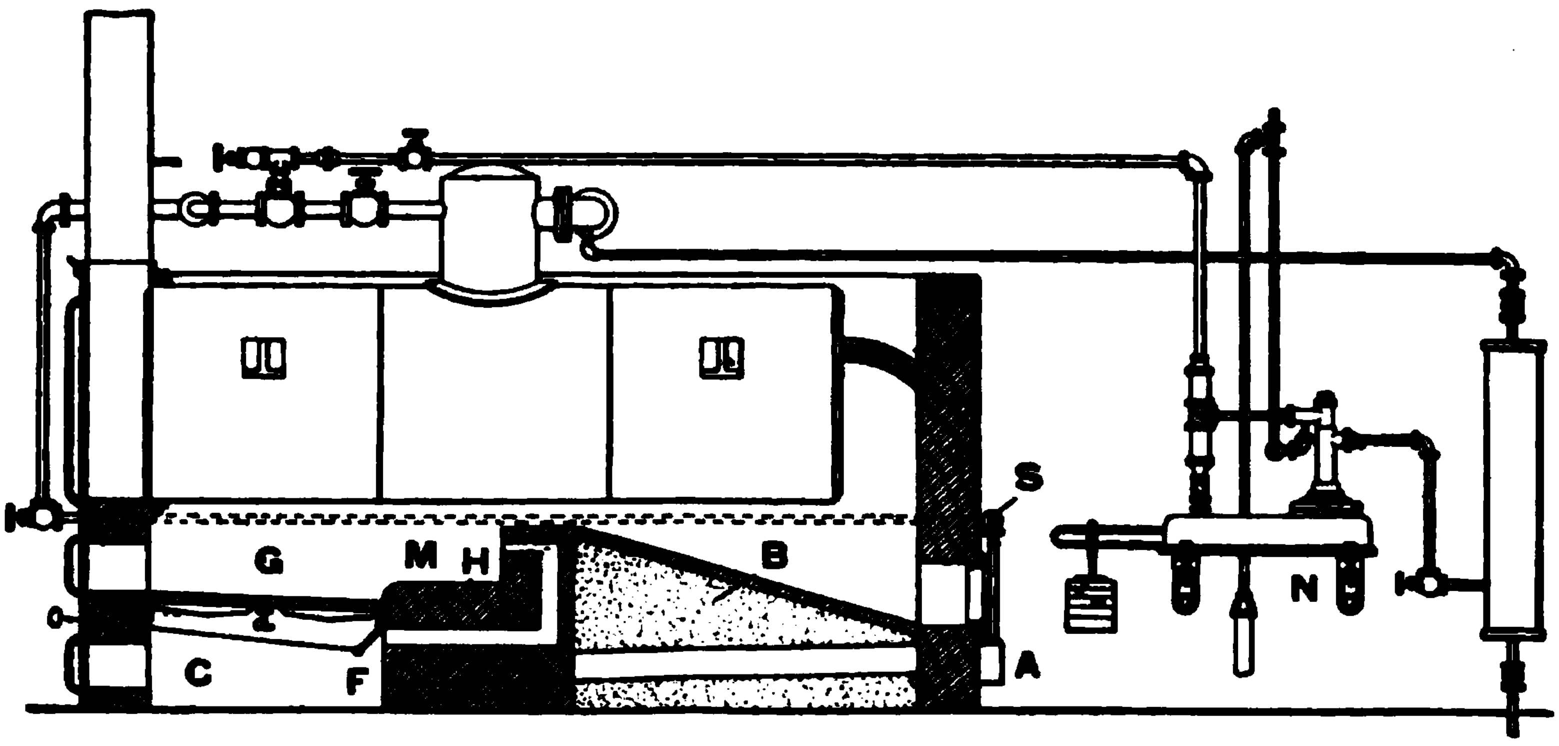


FIG. 129. Parson's Smokeless Furnace:

cated. The bridge wall *H* is provided with an extension platform *M* for holding the unburned fuel when cleaning the fire.

150. Heinrich Smokeless Furnace. — Figs. 130 and 131 show the application of the Heinrich system of forced draft to a return tubular boiler. Hot air is taken from the boiler room above the boilers by a steam jet blower at *A* and forced into the superheating chamber below the combustion chamber. From this chamber part of the air is drawn by the auxiliary blowers *C* and forced through tuyeres above the grate, the rest passing through an opening beneath the bridge wall into the ash pit and up through the bed of fuel. Steam for the blower *A* and the auxiliaries *C* is supplied through an automatic regulator *R*, which opens when the steam pressure falls below the required value. The manufacturers (Heinrich Manufacturing Company, Milwaukee, Wis.) sell this apparatus with a guarantee of 15 per cent saving in fuel over natural draft, common grate bars, and hand firing.

151. Fan Draft. — Fig. 132 shows a typical installation of a centrifugal fan on the forced-draft or plenum principle, the fan creating a pressure in the ash pit and forcing air through the fuel. The most approved method is to pass the air through the bridge wall, thence toward the front of the grate, though it may enter through an underground duct or through the side of the setting. Forced draft is usually adopted in old plants where increased demands for power require that the boilers be forced far above their rating to save the heavy expense

of new boilers, or in plants burning refuse, anthracite culm or screenings, which require an intense draft for efficient combustion. Forced draft is also well adapted for underfeed stokers of the retort type, hollow blast grates, and the closed fire hole system. The air supply may be taken from an air chamber built around the breeching, thereby supplying the heated air to the fan and effecting a lower temperature in the breeching and a higher temperature in the furnace. The objection is sometimes raised against forced draft that the gases tend

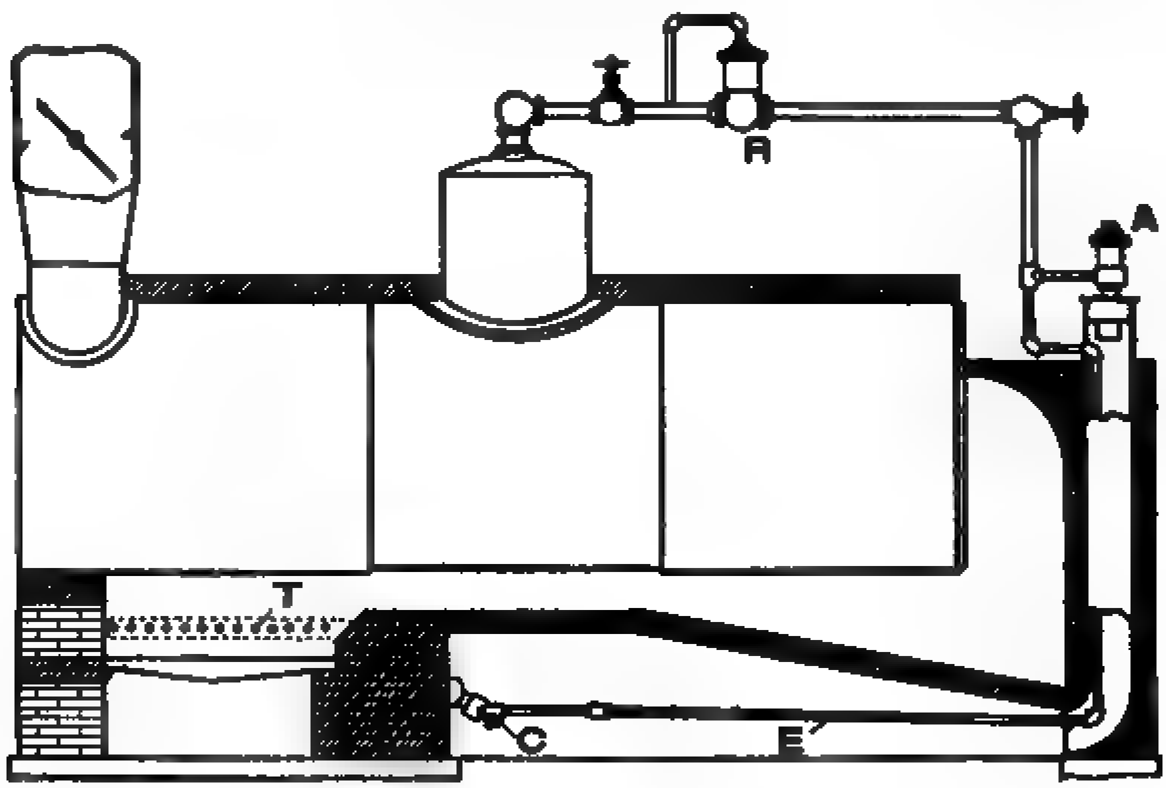


FIG. 130. Heinrich Smokeless Furnace (Sectional Elevation).

FIG. 131. Heinrich Smokeless Furnace (Sectional Plan).

to pass outward through the fire door when the fire is cleaned or replenished, since the pressure in the furnace is greater than atmospheric. This objection may usually be overcome by suitable dampers in the blast pipe which are closed on opening the fire doors. With a boiler plant of 1000 horse power or more the cost of a forced-draft fan, engine, and stack will approximate 20 to 30 per cent of the outlay of an equivalent brick chimney. The power consumption will depend

upon the character and efficiency of the motor or engine and will range from 1 to 5 per cent of the total capacity.

Induced draft as illustrated in Fig. 133 is perhaps the most common substitute for natural draft and is extensively used in street railway and lighting plants which have high peak loads, being ordinarily installed in connection with fuel economizers. The suction side of the fan is connected with the uptake or breeching of the boiler or batteries of boilers and the products of combustion usually exhausted through a stub stack. The illustration shows a typical installation in which two fans of the duplex type are placed above the boiler setting. The fan ducts are generally designed with a by-pass direct to the stack to be used in case of accident or when mechanical draft is not required.

Fig. 132. Typical Forced-Draft System.

Since the fan handles hot gases it must, under the ordinary conditions of practice, have a capacity approximately double that of a forced-draft fan delivering cold air, but the gases being of lower density the power required per cubic foot moved is less.

With forced draft about 300 cubic feet of air are required per pound of coal; with induced draft the fan must handle twice this volume if the gases are exhausted at 500 degrees F. or 450 cubic feet if exhausted at 300 degrees F., a temperature to be expected in connection with economizers.

The advantages of induced draft over forced draft are very pronounced. The pressure in the furnace is less than atmospheric, therefore it is not necessary to shut off the draft in cleaning fires or ash pit, and the fire burns more evenly over the entire grate area and requires less attention than with forced draft. An induced-draft plant costs considerably more than forced draft on account of the larger fan

required, but the operating expenses are but little greater. With a boiler plant of 1000 horse power or more the cost of a single induced-draft fan, engine, stack, etc., will approximate 40 to 50 per cent of the outlay required for a brick chimney of equivalent capacity, and the double-fan outfit will approximate 50 to 60 per cent. The double-fan

FIG. 133. Typical Induced-Draft System.

system is particularly adapted to plants which operate continuously and where even a temporary break-down is a serious inconvenience.

Advantages of Mechanical Draft: Am. Elecn., June, 1898, p. 244, Feb., 1902, p. 63; Eng. Rev., Sept., 1901, p. 4; Eng. Mag., April, 1901, p. 81, March, 1900, p. 931; Elec. Rev., Lond., Feb 3, 1899, p. 186; Cassier's Mag., Nov., 1898, p. 48, Jan., 1905, p. 252, March, 1906; St. Ry. Rev., July 15, 1901, p. 415; Engr. U.S., July 16, 1906, p. 475; Elec. Eng., Aug. 11, 1905, p. 193.

Application of Mechanical Draft to Stationary Boilers: Power, Dec., 1900, p. 30; West. Elecn., Feb. 16, 1901, p. 118; Jour. West. Soc. Engrs., March 19, 1902, p. 271; St. Ry. Rev., July, 1899, p. 463; Cassier's Mag., Nov., 1898, p. 48; Elec. Rev., July 27, 1898, p. 52; Engr. U.S., Jan. 1, 1907.

152. Performance of Fans. — The first satisfactory theory of centrifugal fans was promulgated by Daniel Murgue in 1872. He proved

that theoretically the maximum pressure created by a perfect fan is equivalent to twice the head which would produce a velocity equal to that of the periphery. Thus

$$H = \frac{u^2}{g}, \tag{60}$$

in which

H = maximum difference in pressure in feet of air,
u = peripheral velocity in feet per second, and
g = acceleration of gravity 32.2.

A and B, Fig. 134, represent Pitot tubes inserted in the discharge pipe of a centrifugal blower, A being bent to face the current, while B is at right angles to it. A receives the full impact of the stream, and

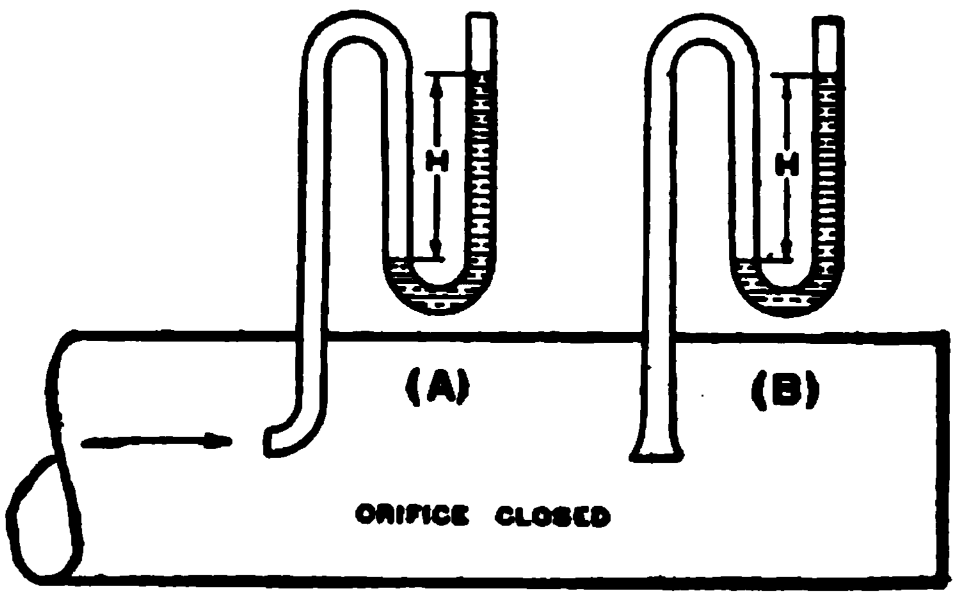


FIG. 134.

the manometer indicates the total pressure, static and velocity, while B registers the static pressure only. With the discharge orifice closed, as in Fig. 134, the velocity becomes zero, and the water depression in both manometers will be the same, due to the static pressure, which, according to Murgue's theory, will be a maximum and, ignoring friction or eddy currents $= \frac{u^2}{g}$.

Example: Determine the maximum pressure, in inches of water, which a perfect fan would exert with discharge orifice closed; diameter of fan 6 feet; r.p.m. 318.

The peripheral velocity is

$$u = 2\pi rn = 6.28 \times 3 \times 318 = 6000 \text{ feet per minute.}$$
$$= 100 \text{ feet per second.}$$

Substituting in Murgue's formula,

$$u = 100 \text{ and } g = 32.2,$$
$$H = \frac{100^2}{32.2} = 310 \text{ feet,}$$

i.e., the pressure created by the fan would be equivalent to the weight of a column of air 310 feet high, or, assuming an air temperature of 75 degrees F., an equivalent head in inches of water of

$$\frac{310 \times 0.074495}{144 \times 0.0361} = 4.45 \text{ inches.}$$

(0.074495 = density of air at 75 degrees F. and 0.0361 = pressure produced by one inch of water in pounds per square inch.)

If the discharge orifice be opened to its maximum (Fig. 135) the static pressure indicated by manometer B becomes zero, since there is no

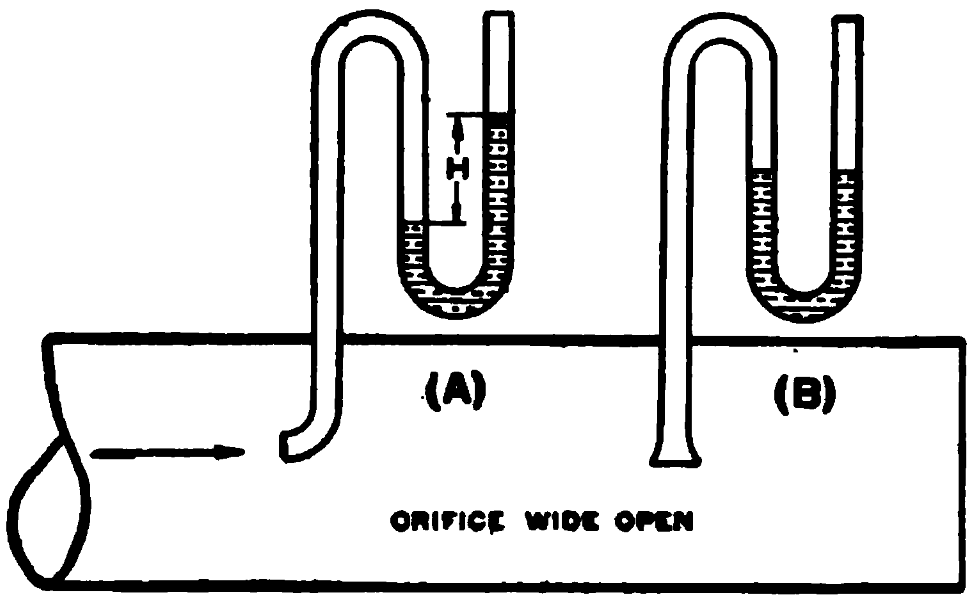


FIG. 135.

resistance due to the air flow, while the water in A stands at a height H the exact equivalent of the velocity head in accordance with the hydraulic formula,

$$v = \sqrt{2\,gH},$$

in which v is the velocity of the air in feet per second.

If the orifice be partially closed, say 50 per cent, as in Fig. 136, B indicates the static pressure, while A gives the dynamic or total pressure due to both velocity and resistance. The difference between A and B is therefore the pressure due to velocity alone. By connecting the two manometers as indicated in Fig. 136 C the velocity pressure is given directly.

PRESSURE. — According to Murgue's theory the maximum pressure which may be developed by a blower or exhauster varies with the square of the speed and may be expressed

$$p = \frac{C\delta u^2}{g},$$

in which

p = pressure, pounds per square foot.
δ = density of the air, pounds per cubic foot.
u = peripheral velocity, feet per second.
C = a coefficient obtained by experiment.

Tables 34 and 35 give the relationship between pressure and speed for various sizes of forced and induced-draft fans.

Fig. 139 shows the relationship between pressure and speed in a 45-inch Buffalo blower as tested at the Armour Institute of Technology.

VELOCITY OF DISCHARGE. — The maximum velocity of the air leaving the tips of the blades varies directly as the peripheral speed,

$$V = Ku, \tag{63}$$

in which

V = velocity of the air discharged, feet per second.
K = a coefficient obtained by experiment.
u = peripheral velocity, feet per second.

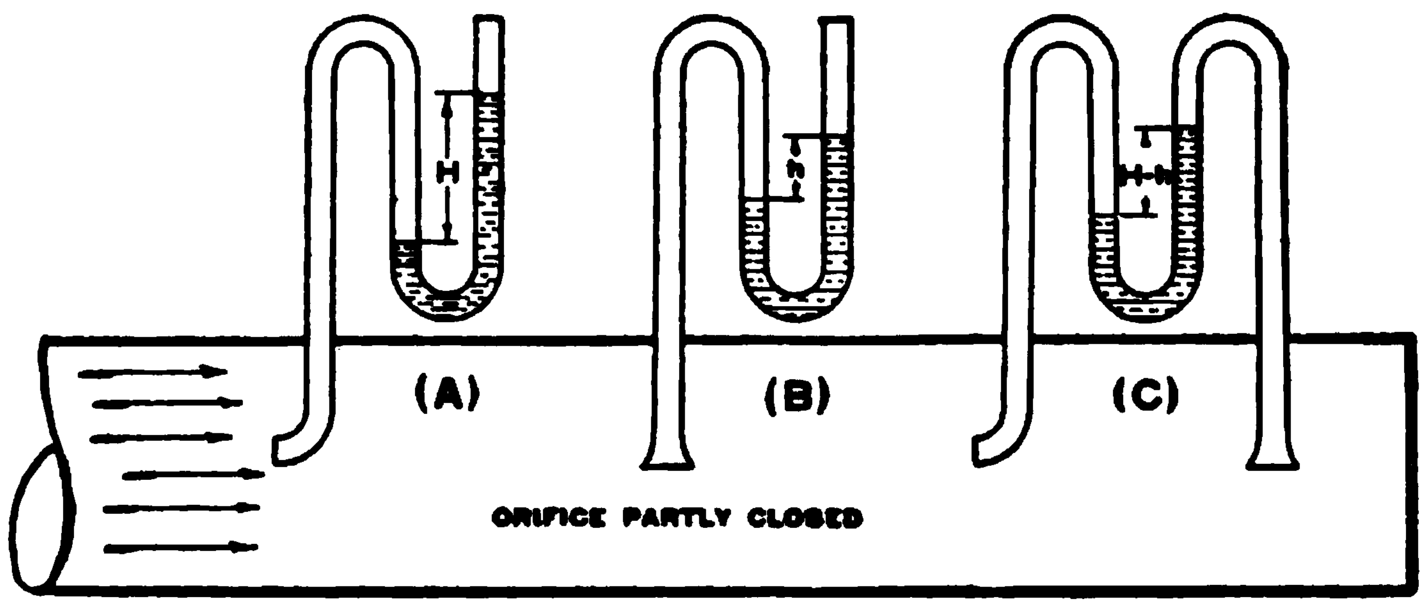


FIG. 136.

For practical purposes the velocity of discharge with outlet wide open may be assumed to be that of the periphery.

CAPACITY. — The relationship between capacity and speed, capacity and discharge opening for a 45-inch pressure blower is given in Figs. 139 and 140.

As will be noted, the capacity varies almost directly with the speed of the wheel and the area of discharge as expressed by the equation

$$Q = B\pi ADN, \tag{63a}$$

in which

Q = cubic feet discharge per minute.
B = coefficient determined from experiment.
A = area discharge opening, square feet.
D = diameter of the wheel.
N = r.p.m. of the wheel.

POWER. — The power required to drive a fan is proportional to the cube of the speed,

$$\text{Horse Power} = XAN^3, \tag{64}$$

in which

X = a coefficient determined by experiment.
A = area discharge outlet, square feet.
N = r.p.m.

The marked increase in power required for even a moderate increase in speed should be borne in mind in selecting a fan. It is as a rule more economical to err in selecting too large a fan than one which must be forced above its rated capacity.

In practice the size of fan is proportioned upon experience rather than theory, the usual procedure necessitating the use of curves based upon the performance of fans of the type under consideration.

The curves in Fig. 137 were computed by Mr. F. R. Still of the American Blower Company, and give the performance of steel-plate fans as manufactured by this company. These curves apply to this type and make of fan only, though the difference is not very great for any type of centrifugal fan. The " ratio of opening " refers to the actual percentage of opening compared with the total discharge. The " ratio of effect " is the relative effect produced by restricting the discharge. The abbreviations are as follows :

D.P. = dynamic or total pressure.
P.V.P. = pressure created by a column of air moving at the same velocity as the periphery.
S.P. = static pressure.
V.P. = velocity pressure = D.P − S.P.

Suppose a fan with an unrestricted inlet and outlet delivers 25,000 cubic feet of air per minute against a head (D.P.) of 0.33 inch with a peripheral velocity requiring 6.16 horse power. It appears from the curves that if the discharge outlet is restricted to 50 per cent of the full area, only 12,500 cubic feet will be delivered; the pressure will be increased to 1.03 inches, and the power required drops to 4.84 horse power. If the outlet be still further reduced to 20 per cent of the full opening the capacity will drop to 5000 cubic feet, the pressure will increase to 1.15 inches, and the power will be decreased to 3.45 horse power. With a discharge area of 60 per cent, the mechanical efficiency is a maximum, and equal to about 43 per cent. With orifice closed the horse power required to drive the fan is about 37 per cent of that required when discharging the maximum volume of air.

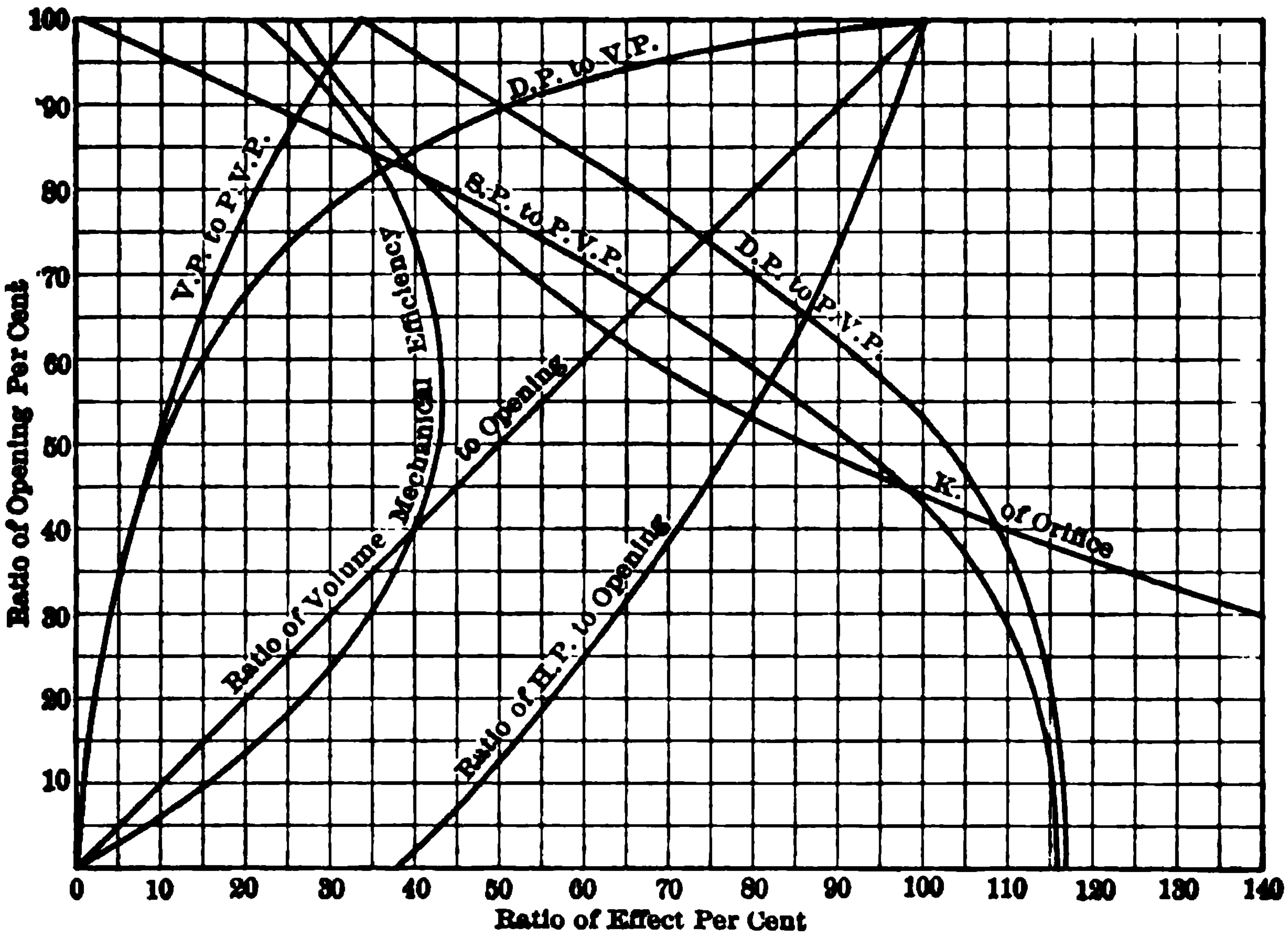


FIG. 137. Performance of Steel Plate Fans.

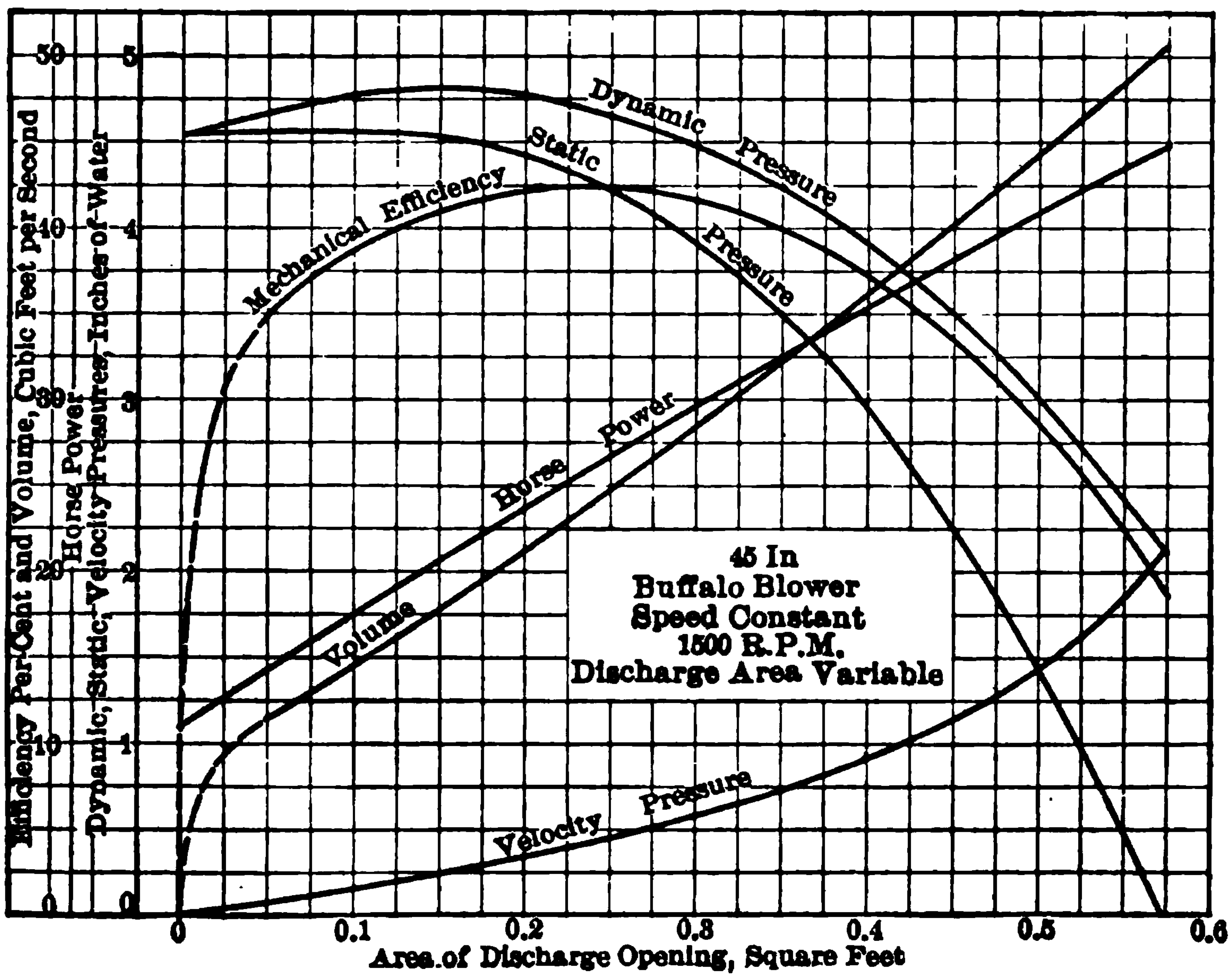


FIG. 138.

Curve " K " in Fig. 137 was determined from the empirical formula (based upon Murgue's theorem)

$$A = \frac{KQ}{\sqrt{P}}, \tag{65}$$

in which

A = area of the inlet orifice, square feet.
Q = volume of gas, thousands of cubic feet per minute.
P = draft at the inlet in inches of water.
K = constant determined by experiment.

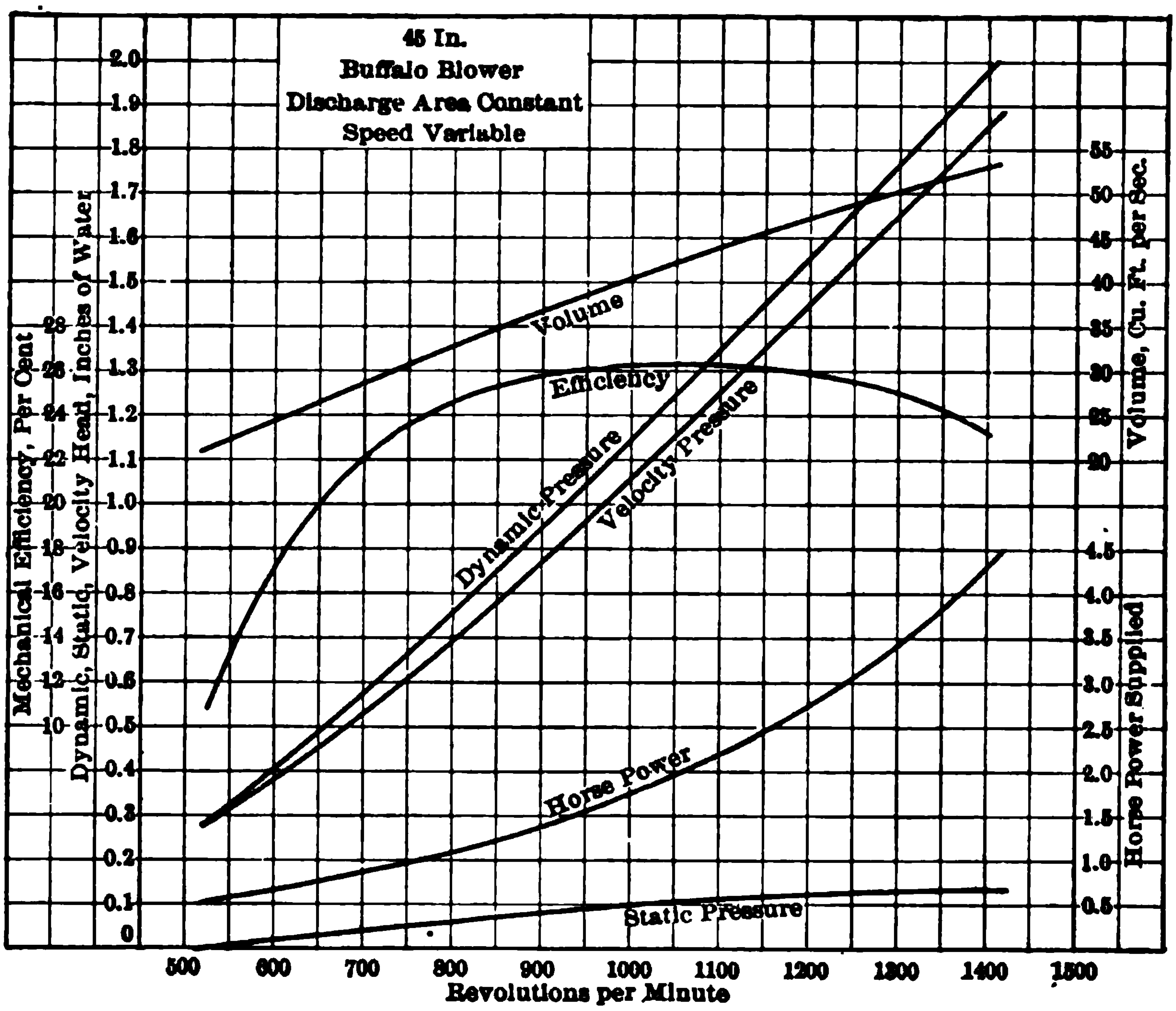


Fig. 139.

The curves in Figs. 138 and 139 are plotted from tests made at the Armour Institute of Technology on a 45-inch Buffalo pressure blower, and are characteristic of this type of fan.

Theory of Fans: Power, May, 1907, p. 287; Engr., Oct. 9, 1903, p. 512; Mach., Aug., 1898; Sib. Jour. of Eng., Nov., 1902; Heat and Vent., Jan. 15, 1897, July, 1899; Prac. Engr., Jan. 16, 1903; Mech. Engr., April 18, 1903; Eng. Rec., Oct. 11, 1902.
Pressure Fans vs. Exhaust Fans: Bulletin Am. Inst. Min. Engrs., Feb., 1909.

153. Determination of Size of Fan.— The following analysis, based upon a paper on Mechanical Draft by F. R. Still of the American

Blower Company, gives a good idea of the usual procedure in determining the size of fan for an induced draft installation. (Jour. West. Soc. Engr., May, 1902.)

Example: Determine the size of induced fan and the approximate power required to drive it, for a boiler plant rated at 1000 horse power; temperature of flue gases 500 degrees F.; heat value of coal 14,000 B.T.U. per pound; ash 5 per cent; draft required, 1 inch of water pressure.

Assuming a boiler efficiency of 70 per cent, the evaporation will be $\frac{14,000}{966} \times 0.70 = 10.15$ pounds of water from and at 212 degrees F. per pound of coal.

Since one boiler horse power is equivalent to the evaporation of 34.5 pounds of water per hour from and at 212 degrees F., the evaporation per hour will be $34.5 \times 1000 = 34,500$ pounds, and the coal burned per hour,

$$\frac{34,500}{10.15} = 3400 \text{ pounds.}$$

Allowing 18 pounds of flue gas per pound of combustible, 5 per cent for ash and 5 per cent for leaks, the fan will have to handle, at 500 degrees F., approximately $20 \times 3400 = 68,000$ pounds of gas per hour, or 26,000 cubic feet per minute. It is customary, when little is known about a plant in which a fan is to be installed, to assume that the resistance is equivalent to restricting the discharge outlet 25 per cent. Hence in this problem the various factors are referred to a "ratio of opening" of 75 per cent (see Fig. 137).

From formula 65, the area of the inlet should be

$$A = \frac{KQ}{\sqrt{P}} = \frac{0.485 \times 26}{1} = 12.6 \text{ square feet,}$$

which corresponds to a diameter of 48 inches. ($K = 0.485$ is taken from the curves in Fig. 127.)

The area of the inlet should not exceed 40 per cent of the area of the side of the wheel; the latter, then, will be

$$\frac{12.6}{0.4} = 31.5 \text{ square feet,}$$

which corresponds to a diameter of 76 inches (6.3 feet).

Referring to Fig. 137, the ratio of dynamic pressure to peripheral velocity pressure (D.P. to P.V.P. at 75 per cent opening) is 0.73. The peripheral velocity pressure will be $\frac{1}{0.73} = 1.37$ inches of water.

The peripheral velocity is

$U = \sqrt{2gH'} = 8.03\sqrt{H'}$, where H' is the peripheral velocity pressure expressed in feet of gas, or,

Since
$$H' = \frac{p \times 62.5}{0.0478 \times 12}, \text{ where } p = \text{inches water,}$$
$$U = 87.5\sqrt{1.37}.$$
$$= 102.5 \text{ feet per second.}$$
$$= 6150 \text{ feet per minute.}$$

The maximum effective discharge area which an inclosed fan of this type may have, and still maintain the pressure equivalent of the peripheral velocity, is usually called the "blast area." With a larger area the pressure will be reduced, but with a smaller area will remain substantially constant. The velocity of the discharge is practically that of the tips of the blades, whence the blast area is equal to $\frac{26{,}000}{6150} = 4.23$ square feet, which, with this type of fan is found to be about $\frac{1}{3}$ the projected rectangle of the wheel, therefore,

The projected rectangle = 4.23 × 3 = 12.7 square feet.

The proper width of periphery is found by dividing this area by the wheel diameter, thus,

$$\text{width of blades} = \frac{12.7}{6.3} = 2.02 \text{ feet} = 24.2 \text{ inches,}$$

and
$$\text{speed of fan} = \frac{6150}{3.14 \times 6.3} = 311 \text{ r.p.m.}$$

$$W = \frac{\text{Volume of gas (cu. ft. per min.)} \times \text{Pressure (lb. per sq. ft.)}}{33{,}000 \times \text{efficiency of fan}}.$$

$$W = \frac{26{,}000 \times 5.2}{33{,}000 \times 0.4} = 10.2 \text{ brake horse power.}$$

(5.2 = pressure in pounds per square foot equivalent to one inch of water, and 0.4 is the mechanical efficiency for 75 per cent opening as taken from curve in Fig. 137.)

Assuming a steam consumption of 70 pounds per brake horse power for a small, simple non-condensing high-speed engine, the steam consumed per hour will be

10.2 × 70 = 714 pounds per hour, or 2.3 per cent of the total steam capacity of the boilers.

Table 34 gives the capacity and horse power required for various sizes of forced-draft fans, and Table 35 gives similar data for induced-draft fans.

154. Chimney *vs.* Mechanical Draft. — The choice of chimney or mechanical draft depends largely upon local conditions. Many power plants with tall stacks are provided with forced-draft apparatus to be used in emergencies, but as a general rule where ordinances require high chimneys mechanical draft is not considered. In a few isolated cases stokers of the forced-draft type are used in connection with chimneys as high as 250 feet, but such installations are rare, and not to be recommended.

Where there are no limitations to the height of stack, mechanical draft offers many advantages over chimney draft, especially for railroad work and large lighting plants. With certain types of grates and for low-grade fuels and anthracite culm or dust, it is indispensable. Again, where a fair quality of fuel is obtainable the size of plant may determine the choice.

First Cost: In small plants of say 100 to 150 horse power the cost of a guyed steel chimney, 75 feet in height or less, would be considerably less than that of a mechanical-draft system, and once erected cost practically nothing for operation, while the power required to operate a fan in so small a plant would amount to 5 per cent or more of the total steaming capacity.

TABLE 34.

CAPACITIES OF FORCED-DRAFT FANS.

(Power.)

For Forced Draft, Temperature of Air 60°.

Diameter of Fan.	Cubic Feet of Air Delivered to Furnace per Minute.	Pressure in Inches of Water. 0.5		0.75		1.00		1.25		1.50		2.00		2.50	
		R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.
2′ 6″	4,200	510	1.6	560	1.8	600	1.9	640	2.1	710	2.3	780	2.5	850	2.7
3′	5,800	430	2.2	460	2.4	490	2.6	530	2.8	590	3.1	640	3.4	710	3.8
3′ 6″	7,800	360	3.0	400	3.3	420	3.5	450	3.8	500	4.2	550	4.6	610	5.1
4′	10,000	320	3.9	350	4.2	370	4.4	400	4.9	440	5.4	480	5.9	530	6.5
4′ 6″	12,400	290	4.8	310	5.2	330	5.6	360	6.0	400	6.7	430	7.3	470	8.0
5′	15,200	250	5.9	270	6.4	290	6.8	310	7.4	350	8.2	380	8.9	420	9.8
5′ 6″	18,200	230	7.0	250	7.7	270	8.2	300	8.8	330	9.8	360	10.6	390	11.8
6′	21,400	210	8.3	230	9.1	250	9.6	260	10.4	290	11.5	320	12.5	350	13.9
7′	28,800	180	11.2	200	12.2	210	13.0	230	14.0	250	15.5	280	16.8	300	18.7
8′	37,200	160	14.4	170	15.7	190	16.7	200	18.1	220	20.1	240	21.8	270	22.5
9′	46,800	140	18.1	160	19.8	170	21.1	180	22.7	200	25.3	220	27.4	240	30.3
10′	57,400	130	22.2	140	24.3	150	25.8	160	27.9	180	3.1	200	33.6	210	37.2

Discharge velocity 2000 feet per minute.

TABLE 35.

CAPACITIES OF INDUCED-DRAFT FANS.

(*Power.*)

For Induced Draft, Temp. of Flue Gases 500°.

Diameter of Fan.	Cubic Feet of Air at 60° Temp. Drawn into Furnace per Minute.	Pressure in Inches of Water.													
		0.5		0.75		1.00		1.25		1.50		2.00		2.50	
		R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.
2′ 6″	3,000	688	2.2	756	2.4	810	2.6	864	2.8	958	3.1	1053	3.4	1147	3.6
3′	4,200	580	3.0	621	3.2	661	3.5	715	3.8	796	4.2	864	4.6	958	5.1
3′ 6″	5,700	486	4.0	540	4.5	567	4.7	607	5.1	675	5.7	742	6.2	823	6.9
4′	7,300	432	5.3	472	5.7	500	6.1	540	6.6	594	7.3	648	8.0	715	8.8
4′ 6″	9,300	390	6.5	418	7.0	445	7.5	486	8.1	540	9.0	580	9.8	634	10.8
5′	11,100	337	8.0	364	8.6	391	9.2	418	10.0	472	11.1	513	12.0	567	13.2
5′ 6″	13,300	310	9.5	337	10.4	364	11.1	405	11.9	445	13.2	486	14.3	526	15.9
6′	15,600	283	11.2	310	12.3	337	13.0	351	14.0	391	15.5	432	16.9	472	18.7
7′	21,000	243	15.1	270	16.5	283	17.5	310	18.9	337	20.9	378	22.6	405	25.2
8′	27,100	216	19.4	230	21.2	256	22.5	270	24.4	297	27.1	324	29.4	364	30.4
9′	34,200	189	24.4	216	26.7	230	28.5	243	30.6	270	34.1	297	37.0	324	40.9
10′	41,900	175	30.0	190	32.8	202	34.8	216	37.6	243	41.8	270	45.3	283	50.2

A tall, self-supporting chimney for larger plants, however, is very costly as compared with a fan system of equal capacity. For example, a brick chimney 175 feet high and 10 feet in diameter, foundation and all, capable of furnishing the necessary draft for a 3000-horse-power plant, will cost about $10,000. A two-fan induced system of equivalent capacity will cost in the neighborhood of $5000, a one-fan system $3500, and a forced-draft system $2500. See Fig. 140. With interest at 5 per cent, depreciation 5 per cent, taxes 1 per cent, and insurance one-half per cent, the annual fixed charges will be $575, $402.50, $287.50, respectively, for the fan equipment.

Depreciation and Maintenance : The depreciation of a well-designed masonry or concrete stack is very low, and 2 per cent is a liberal factor. Maintenance is practically negligible, as it requires no attention whatever for years. A steel stack, however, must be kept well painted or corrosion will take place rapidly. The depreciation and maintenance charges on a mechanical-draft system will range from 4 per cent to 10 per cent of the original outlay.

Cost of Operation: Once erected, the comparative cost of operating a chimney is practically nothing; that is, of course, on the assumption that the chimney and fan exhaust equal volumes of gas per pound of

fuel and at the same temperature. A fan system requires for its operation from one and one-half per cent to five per cent of the total steaming capacity of the plant, depending upon the type and character of the fan engine or motor, and the conditions of operation.

Efficiency: With fan draft a very thick fire can be maintained on the grate, thus permitting a high rate of combustion, and minimum air per pound of fuel, both of which result in increased boiler efficiency.

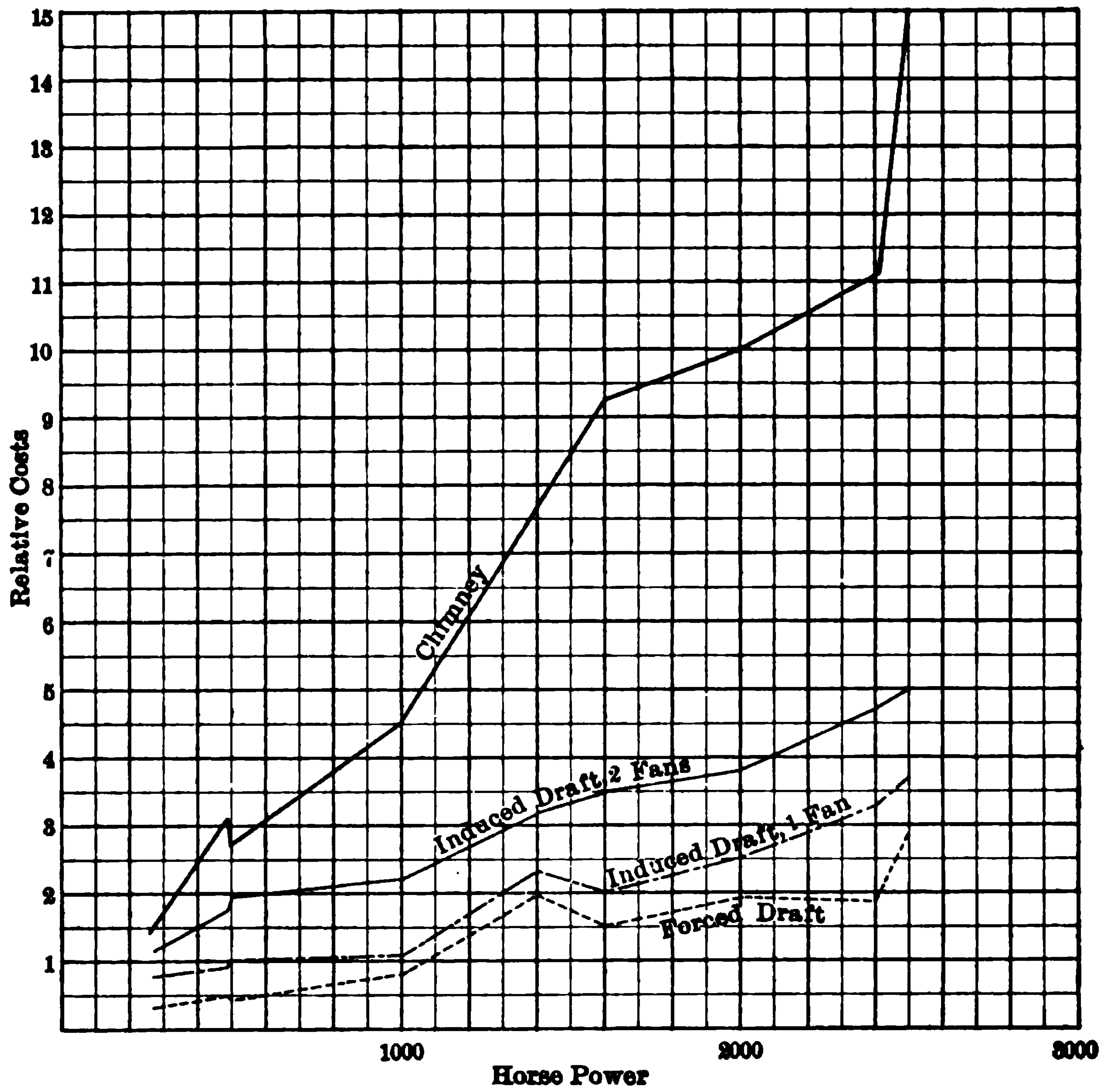


FIG. 140. Comparative Costs of Chimneys and Mechanical Draft. (W. B. Snow.)

The influence of the rate of combustion on air supply is illustrated in Fig. 141. For the same temperature of discharge each pound of air in excess of theoretical requirements results in a loss of about one per cent of the total heat in the fuel. (See Table 3.) With fan draft an average figure is 18 pounds of air per pound of bituminous coal against 24 pounds for the chimney, a saving of 5 per cent in favor of the fan. Again, a fan permits of a low temperature of the flue gases without

affecting the draft, while lowering the temperature in the chimney reduces the draft as shown in Table 24. From Table 4 we see that a reduction in flue gas temperature of 25 degrees F. will increase the boiler efficiency about one per cent. With an economizer the flue gases may be reduced to 350 degrees F., with a net saving of about $500 - 350 = 150$, or 6 per cent of the total fuel. It is in this connection that the fan draft is peculiarly suitable. Of course, the chimney may be provided with an economizer, effecting the same reduction in tem-

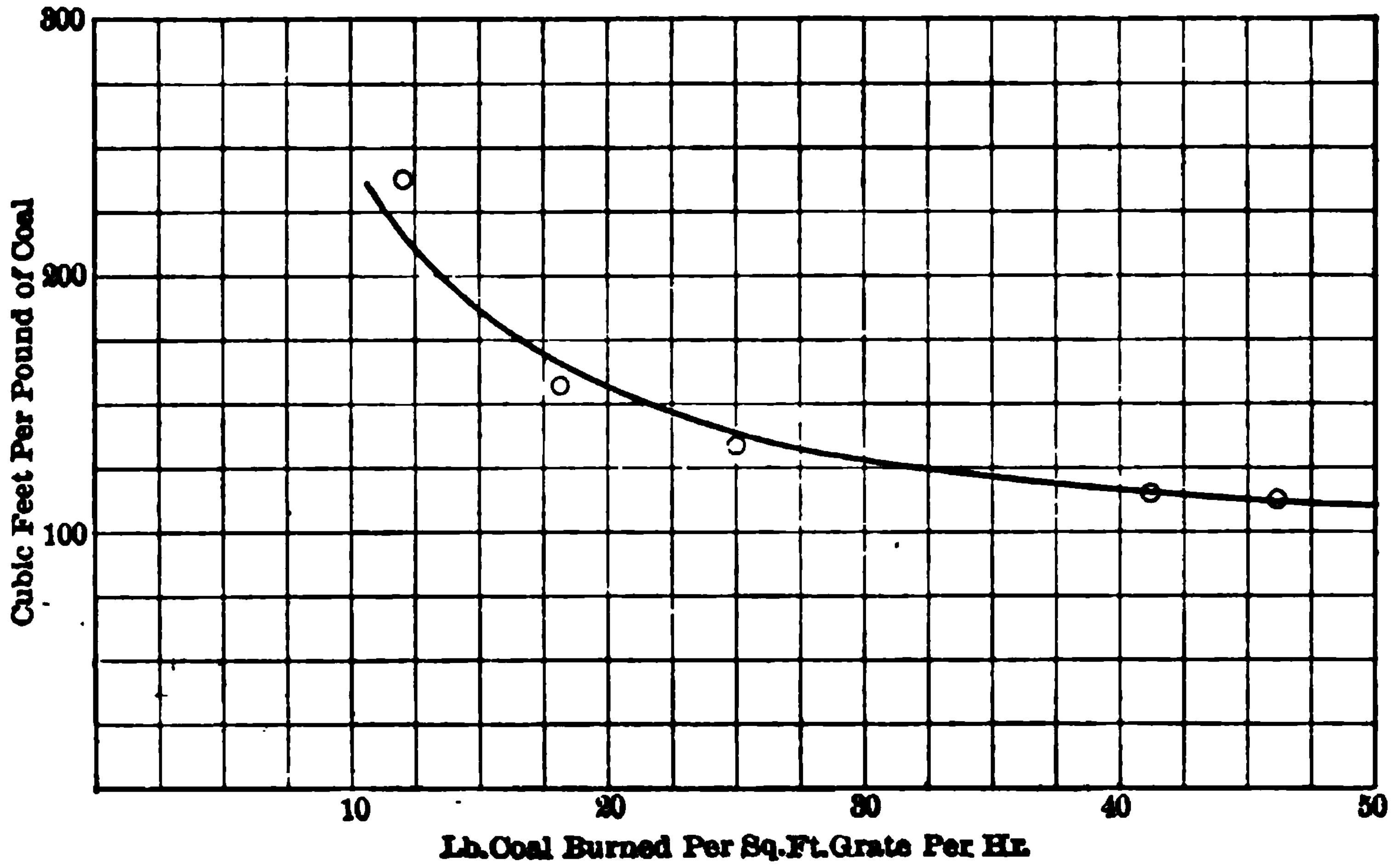


FIG. 141. Influence of Rate of Combustion on Air Supply. — Forced Draft.

perature, but its height must be made sufficiently great to overcome the additional resistance of the economizer and the reduction in temperature of the chimney gases.

Flexibility: With a fan the draft may be readily regulated for sudden increased or decreased requirements, independent of the boiler performance. Damp and muggy days appreciably affect the draft of a chimney, as do adverse air currents and high winds.

Smoke: Smokeless combustion is more readily effected with artificial draft than with natural draft, as a thicker fire can be carried, and the correct proportion of air can be more readily adjusted.

Comparative Tests of Chimney and Mechanical Draft: Power, July, 1901, p. 22; Eng. Rec., July 25, 1903, p. 102; Eng., U. S., May 1, 1899, p. 105; Engr., U. S., April 15, 1907.

155. Balanced Draft. — Fig. 142 illustrates an application of the McLean "Balanced Draft" system to a water-tube boiler. The equipment consists of a blower, the speed of which is regulated

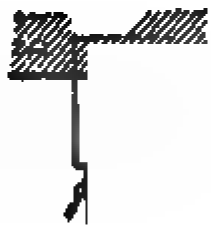


Fig. 142. "Balanced Draft" System.

by the steam pressure, so that the draft in the fire box is maintained at approximately atmospheric pressure. The chief claims for this system are (1) the velocity of the gases over the tubes is reduced, and short circuiting is prevented; (2) the correct proportion of air to fuel is readily maintained; (3) infiltration of air through the setting is impossible, as the pressures are "balanced"; (4) sudden changes in load are correctly taken care of. Tests of the apparatus at the Fuller Building, New York, gave excellent results (Trans. A.S.M.E., 26-641).

CHAPTER IX.

STEAM ENGINES.

156. Introductory. — The reciprocating steam engine is the most widely distributed and generally adopted prime mover in the power world although its field of usefulness has been greatly encroached upon in recent years by the steam turbine and the gas engine. The steam turbine has practically superseded the piston engine for large steam electric plants, while in other fields the gas engine offers many advantages, but the reciprocating steam engine is still an important heat engine and will probably continue to be a factor in the power world for years to come.

The type of engine best suited for a given installation is the one which delivers the required power at the lowest cost, measured in dollars and cents, taking into consideration interest on the investment, operating expenses, maintenance and depreciation.

157. Ideal Engine. — The thermal efficiency of the steam engine is expressed by the ratio of the heat equivalent of the work done on the piston per unit of time, to the heat supplied. The degree of perfection realized is ascertained by comparing the performance of the real engine with that of an ideally perfect engine, working between the same temperature limits. The theoretical limit of perfection is that defined by the Carnot cycle, the efficiency of which is represented by the equation

$$E_e = \frac{T_1 - T_2}{T_1}, \tag{66}$$

in which

T_1 = the highest absolute temperature of the working fluid.
T_2 = the lowest absolute temperature of the working fluid.

The upper limit of temperature is that corresponding to boiler pressure, and the lower limit to that of the exhaust steam. Evidently the greater the temperature range the more nearly does the ideal efficiency approach unity, but with the present limits of temperature used in steam engines, it cannot exceed about 35 per cent.

The nearest approach of any actual engine to the Carnot cycle is accomplished by the Nordburg system of progressive feed heating,

in which the feed water is successively heated from the receivers intermediate between each pair of cylinders. (*Engineering News*, May 4, 1899, p. 283.) Table 36 gives the Carnot efficiencies of condensing and non-condensing engines for ordinary ranges of steam pressures.

The Carnot cycle is theoretically impossible for an engine using superheated steam at constant pressure, and, in general, it is not very closely simulated by engines using saturated steam. It is, therefore, more instructive to select an ideal cycle which more nearly represents the performance of the actual engine. The diagram representing the operation of this perfect engine is shown in Fig. 142a, and is called the *non-conducting* or *Rankine* cycle. *ab* represents the admission of dry steam from the boiler at pressure p_1; *bc* is an adiabatic expansion to exhaust pressure p_2; *cd* represents the exhaust, and *da* is an adiabatic compression to the initial pressure.

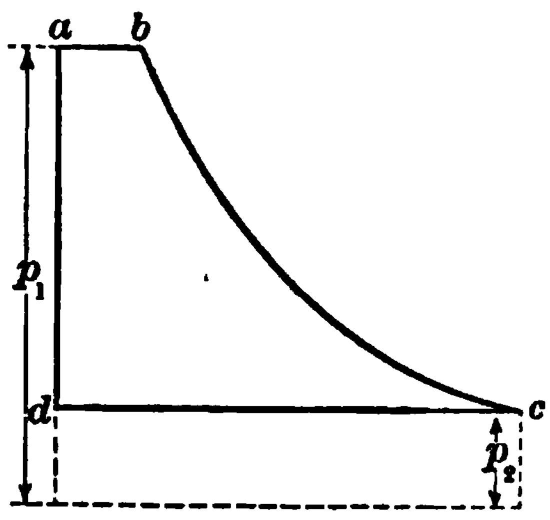


FIG. 142a.

The heat necessary to raise the feed water from the temperature of exhaust, or *ideal feed water temperature*, to the temperature in the boiler and evaporate it into dry steam is

$$H_1 = r_1 + q_1 - q_2, \tag{66}$$

in which

H_1 = quantity of heat supplied to the cylinder per pound of steam.

r_1 = heat of vaporization at pressure p_1.

q_1 = heat of the liquid at pressure p_1.

q_2 = heat of the liquid at pressure p_2.

The heat, H_2, exhausted from the cylinder and which must be withdrawn when it is condensed is

$$H_2 = x_2 r_2, \tag{66a}$$

in which

x_2 = quality of the steam at pressure p_2.

r_2 = heat of vaporization at pressure p_2.

x_2 may be calculated by the aid of equation

$$x_2 = \frac{T_2}{r_2}\left(\frac{r_1}{T_1} + \theta_1 - \theta_2\right), \tag{66b}$$

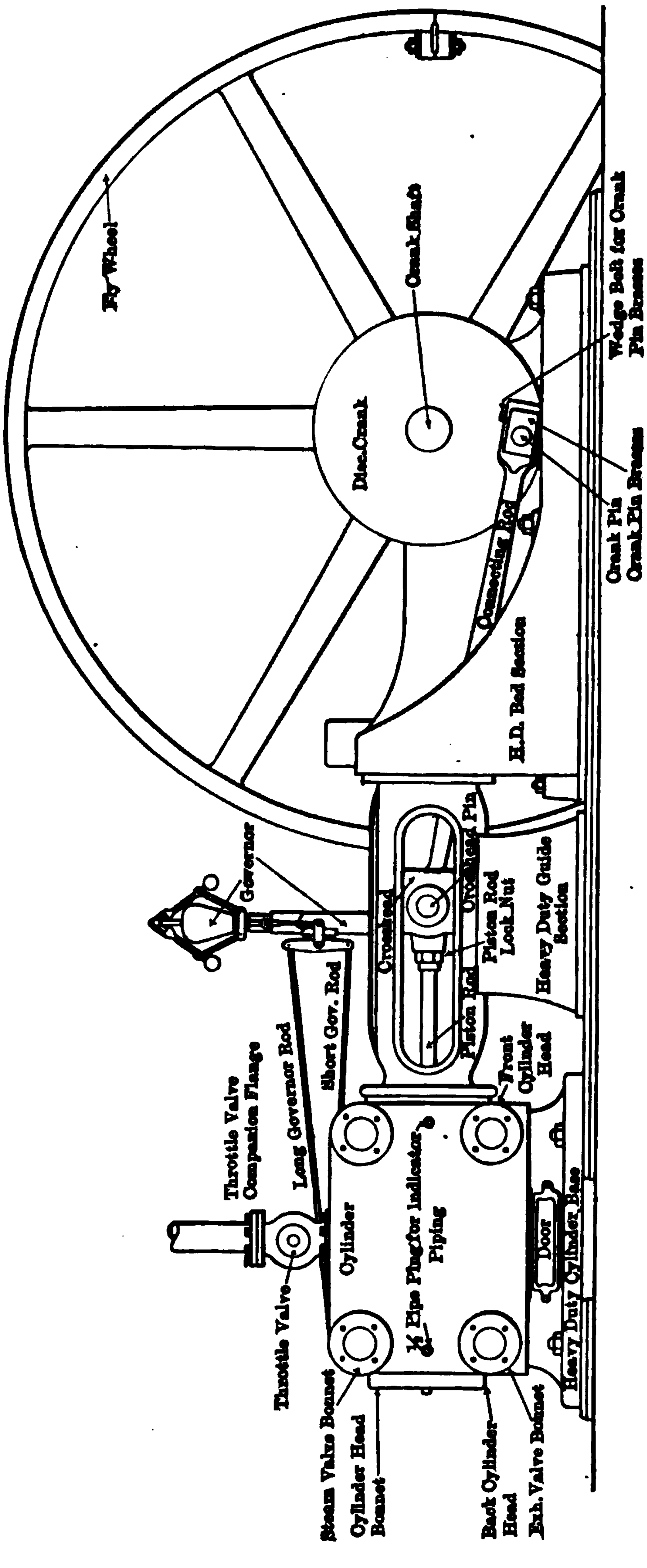


FIG. 142b. Side Elevation of a Typical Corliss Engine, Single Cylinder, Belt Drive.

FIG. 142c Plan View of a Typical Corliss Engine, Single Cylinder, Belt Drive.

in which

T_2 = absolute temperature of steam at pressure p_2.
T_1 = absolute temperature of steam at pressure p_1.
θ_1 = entropy of the liquid at pressure p_1.
θ_2 = entropy of the liquid at pressure p_2.

Other notations as above.

The heat changed into work per pound of steam is

$$H_1 - H_2 = r_1 + q_1 - q_2 - x_2r_2, \tag{67}$$

and the efficiency, E_r, of the cycle is

$$E_r = \frac{H_1 - H_2}{H_1} = \frac{r_1 + q_1 - x_2r_2 - q_2}{r_1 + q_1 - q_2}. \tag{67a}$$

The steam consumption W, or water rate, lbs. per h.p. hr. of the perfect engine, may be expressed

$$W = \frac{2545}{r_1 + q_1 - q_2}. \tag{67b}$$

If the steam entering the cylinder is wet and of quality x_1, substitute x_1r_1 in above equations for r_1.

If the steam is superheated at admission but becomes moist at the lower pressures, which is the usual case, the efficiency may be expressed

$$E_r = \frac{r_1 + q_1 + c_1t_s - x_2r_2 - q_2}{r_1 + q_1 + c_1t_s - q_2}, \tag{67c}$$

in which

c_1 = *mean* specific heat of the superheated steam at pressure p_1.
t_s = *degree* of superheat or difference in temperature between the superheated and saturated steam at pressure p_1.

x_2 may be calculated by the aid of equation

$$\int_{T_1}^{T} \frac{c\,dt}{T} + \frac{r_1}{T_1} + \theta_1 = \frac{x_2r_2}{T_2} + \theta_2, \tag{67d}$$

in which

t and T = thermometric and absolute temperatures of the superheated steam.
c = *true* specific heat of superheated steam at temperature t.

Other notations as above.

FIG. 142d. Example of a Modern Piston Engine Power Plant. Western Electric Company, Chicago.

For many purposes equation (67d) may be expressed

$$c_1 \log_e \frac{T_1}{T_s} + \frac{r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2. \tag{67e}$$

For highly superheated steam in which the steam is still superheated at exhaust

$$E_r = \frac{r_1 + q_1 + c_1 t_s - r_2 - q_2 - c_2 t_s'}{r_1 + q_1 + c_1 t_s - q_2}, \tag{67f}$$

in which

c_2 = *mean* specific heat of the superheated steam at exhaust.

t_s' = degree of superheat at exhaust.

t_s' may be calculated by the aid of equation

$$\int_{T_1}^{T} \frac{c\,dt}{T} + \frac{r_1}{T_1} + \theta_1 = \int_{T_2}^{T_2'} \frac{c'dt}{T} + \frac{r_2}{T_2} + \theta_2. \tag{67g}$$

Problems connected with the Rankine cycle may be conveniently solved by temperature-entropy tables to be found in connection with the usual steam tables or by the Mollier diagram as described in Appendix H.

158. Thermal Efficiency of the Actual Engine. — In calculating the thermal efficiency of the real engine the heat supplied is reckoned above the sensible heat of the exhaust, thus,

$$E_t = \frac{\text{Heat converted into useful work}}{\text{Heat supplied}} \tag{68}$$

$$= \frac{42.42}{\text{B.T.U. supplied per I.H.P. per minute}} \tag{69}$$

$$= \frac{42.42}{w\,(x_1 r_1 + q_1 - q_2)}, \tag{70}$$

in which

w = the weight of steam supplied to the engine per indicated horse power per minute, or per brake horse power per minute, depending upon whether the efficiency is to be referred to the indicated or to the brake horse power of the engine.

Other notations as in (66) and (67).

The figure obtained by dividing the efficiency of the real engine by that of the ideal engine is called the *efficiency ratio,* and is a measure of the extent to which the theoretical possibilities are realized.

The efficiency ratio is calculated on the basis of the indicated horse power or the developed horse power:

$$\text{Eff. Ratio} = \frac{E_t}{E_r} = \frac{42.42}{w\,(H_1 - H_2)}. \tag{71}$$

This ratio expressed in different terms has been referred to as "Potential Efficiency" by C. V. Kerr (Trans. A.S.M.E., 25-920), and as "Cylinder Efficiency" by Professor Reeve.

The commercial economy of an engine is measured by the cost of producing power, and does not necessarily depend upon its thermal efficiency. The performances of steam engines are frequently stated in terms of (*a*) pounds of steam utilized per horse-power hour, (*b*) pounds of coal per horse-power hour, (*c*) cost in cents per horse-power hour, (*d*) B.T.U. per horse-power hour. From a commercial stand-

TABLE 36.

STEAM-ENGINE EFFICIENCIES.

(Saturated Steam.)

Gauge Press.	Non-Condensing: Back Pressure 14.7 Absolute.				Condensing: Back Pressure 1 Pound Absolute.			
	Carnot Cycle.	Rankine Cycle (a).	Actual * (b).	Ratio $\frac{b}{a}$.	Carnot Cycle.	Rankine Cycle (a).	Actual (b).	Ratio $\frac{b}{a}$.
				%				%
25	7.5	7.3	5.5	76.0	22.6	21.0	11.6	55.0
50	11.2	10.7	8.5	80	25.7	23.5	13.5	60
75	13.7	13.0	10.4	80	27.8	25.3	15.9	61
100	15.7	14.8	12.0	81	29.5	26.7	20.2	76
125	17.3	16.3	13.5	83	30.8	27.8	20.3	74
150	18.7	17.5	14.3	82	32.0	28.8	21.6	75
175	19.8	18.5	14.8	80	32.9	29.6	21.9	74
200	20.8	19.3	15.2	79	33.7	30.2	22.6	75
225	21.6	19.9	15.5	78	34.5	30.6	22.6	74
250	22.4	20.5	....	......	35.1	31.0	....	......
275	23.0	21.0	....	......	35.6	31.3	....	......
300	23.6	21.4	....	......	36.0	31.5	....	......

* Best recorded performance of the actual engine, 1907.

point the cost of producing power is the most important basis of comparison, but the latter expression is most satisfactory for scientific purposes, since it gives a basis of comparing the performances of all types of heat engines.

159. Mechanical Efficiency. — The power of an engine may be expressed in terms of indicated horse power, brake horse power, or pump horse power, according to the class of engine. The ratio of the brake to the indicated power is the mechanical efficiency of the engine, the ratio of the electric horse power to the indicated power is the mechanical efficiency of the engine and generator combined, and the ratio of the pump horse power to the indicated power of the steam engine is the mechanical efficiency of the engine and pump

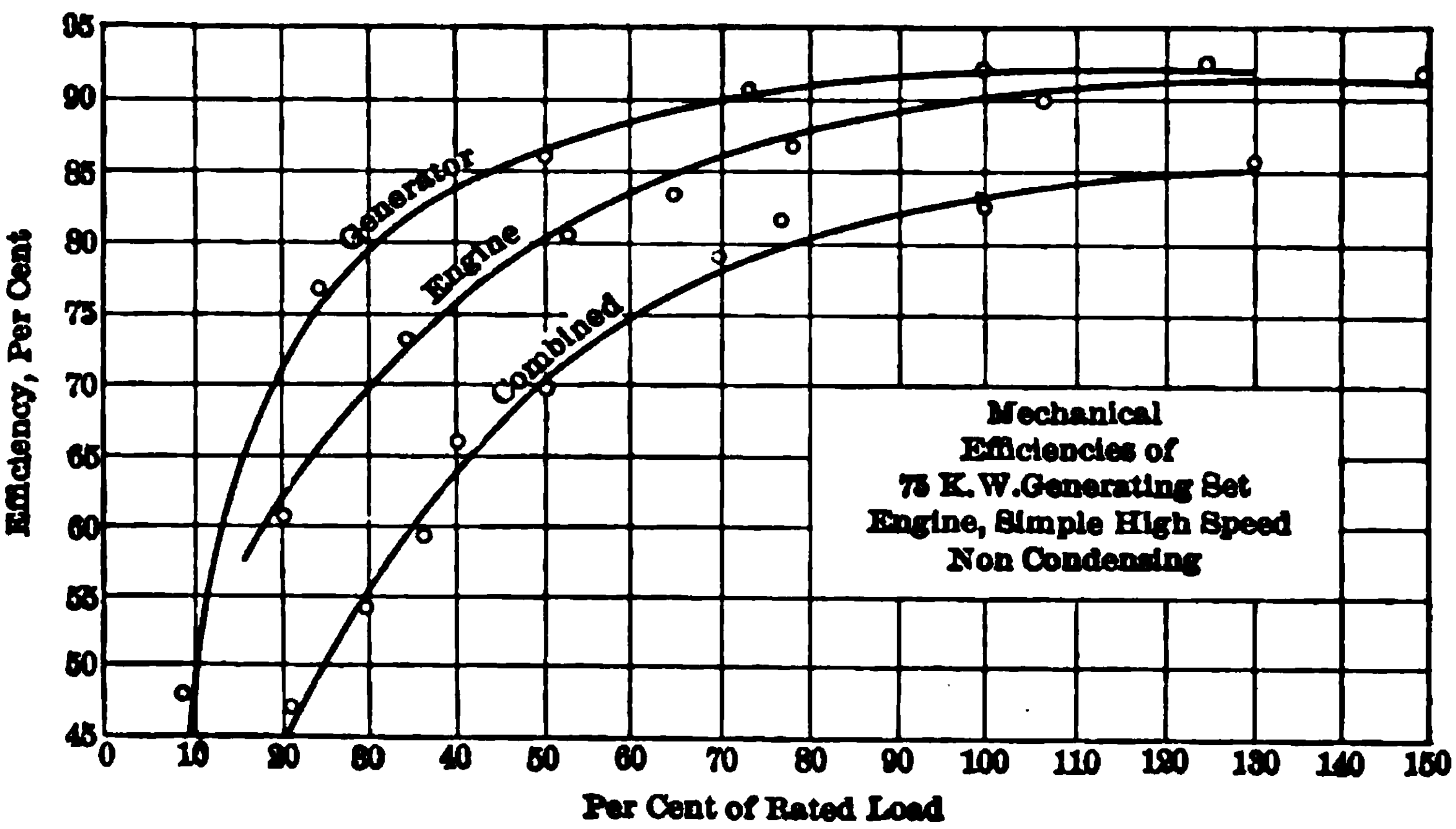


FIG. 143.

combined. The percentage of work lost in friction is therefore the difference between 100 per cent and the mechanical efficiency.

Table 37 shows the mechanical efficiencies for several types of engines, and Fig. 143 the combined efficiency of a direct-connected high-speed engine and generator. (See Engine Friction, par. 167.)

Mechanical Efficiency: Peabody, Thermodynamics, p. 430; Spangler, Steam Engineering, p. 205; Ripper, Steam Engine, p. 275; Ewing, Steam Engine, p. 186.

The following numerical example will illustrate the calculation of the various efficiencies mentioned:

A simple high-speed engine uses 30 pounds of steam per I.H.P. hour; initial pressure 100 pounds per square inch, gauge; exhaust pressure, atmospheric; I.H.P., 120; D.H.P., 102. Steam assumed to be dry and saturated at throttle. Required: (1) the actual thermal efficiency; (2) the efficiency of the Carnot cycle; (3) the efficiency of the Rankine cycle; (4) the efficiency ratio; (5) the mechanical efficiency.

TABLE 37.

MECHANICAL EFFICIENCIES OF ENGINES.

Kind of Engine.	Horse Power.	Efficiency at Full Load.
Simple:		
1. High-speed, non-condensing	150	95.5
2. High-speed, condensing	170	96
3. Low-speed, non-condensing	275	94
4. Low-speed, condensing	....	....
Compound:		
5. High-speed, non-condensing	150	94
6. High-speed, condensing	160	98
7. Low-speed, non-condensing	900	95
8. Low-speed, condensing	1000	95
Triple: (combined efficiency of engine and pump)		
9. Pumping engine	865	97.4
Quadruple: (combined efficiency of engine and pump)		
10. Pumping engine	712	93

1. Buffalo Simple engine, 12 × 12, Elec. World, Sept., 1904, p. 147.
2. Reeves Simple engine, 15 × 14, Elec. World, Oct. 1, 1904, p. 587.
3. 24 × 48 Hamilton Corliss at Armour Inst. of Tech., 1898.
4.
5. Reeves Compound; Eng. Rec., July 1, 1905, p. 24.
6. Reeves Compound; Eng. Rec.
7. 21, 41 × 30 Cross Compound Ball & Wood, West Albany Station, N.Y.C. & H.R.R.
8. 20, 40 × 42 Rice & Sargent; A.S.M.E., 29-1276.
9. Allis Pumping Engine; Power, May, 1906, p. 299.
10. Nordburg Pumping Engine; Eng. News, May 4, 1899, p. 280.

(1) The actual thermal efficiency is

$$E = \frac{42.42}{W\,(x_1r_1 + q_1 - q_2)}$$
$$= \frac{42.42}{30/60\,(879.6 + 308.9 - 180.3)}$$
$$= 0.084.$$

That is, only 8.45 per cent of the heat supplied to the cylinder above the temperature of the exhaust steam is converted into work. The assumption that the exhaust steam may be used to heat the feed water to its own temperature justifies reckoning the heat supplied above the exhaust temperature.

(2) The efficiency of the Carnot cycle is

$$E_c = \frac{T_1 - T_2}{T_1}$$
$$= \frac{798.8 - 673}{798.8} = 0.157,$$

which means that if the engine were a perfect one employing the Carnot cycle between the same extremes of temperature as the actual engine, 15.7 per cent of the energy supplied would be converted into useful work.

(3) On the basis of the Rankine cycle the ideal efficiency becomes

$$E_r = \frac{x_1r_1 + q_1 - x_2r_2 - q_2}{x_1r_1 + q_1 - q_2}$$
$$= \frac{879.6 + 308.9 - 0.885 \times 969.7 - 180.3}{879.6 + 308.9 - 180.3}$$
$$= 0.149,$$

the numerator representing the heat converted into work per pound of steam and the denominator the heat supplied. Thus if the engine be

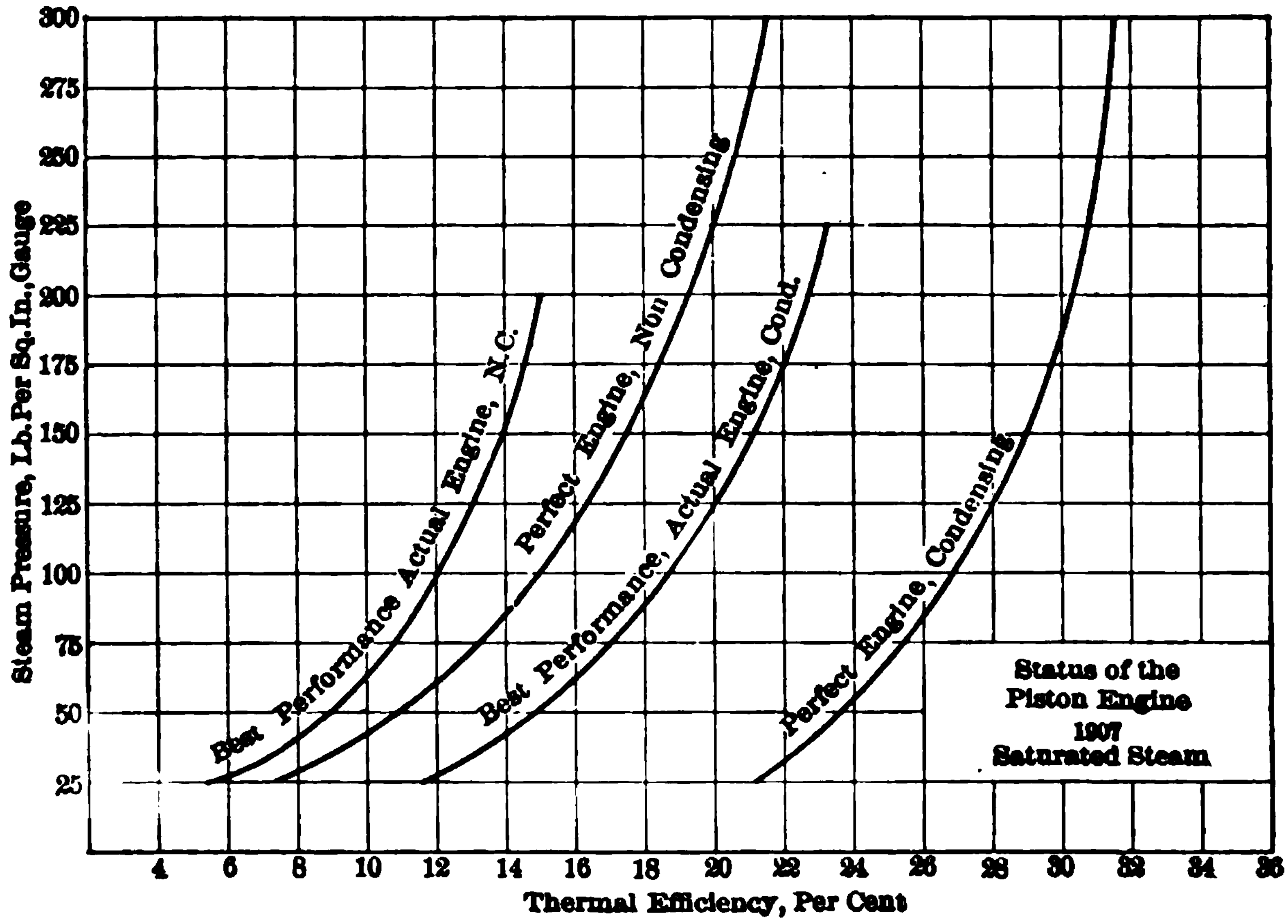


FIG. 144.

assumed to have a non-conducting cylinder and work on the Rankine cycle, it would be capable of utilizing 14.9 per cent of the heat supplied.

(4) The efficiency ratio is expressed:

$$\text{Efficiency ratio} = \frac{E_t}{E_r} = \frac{8.45}{14.9} = 56.7 \text{ per cent,}$$

which indicates the degree of perfection of the engine or the extent to which it realizes the maximum efficiency theoretically possible. The weight of steam which would have to be consumed per horse power per hour to actually attain the Rankine efficiency would be

$$30 \times \frac{0.0845}{0.149} = 30 \times 0.567 = 17 \text{ pounds.}$$

(5) The mechanical efficiency $= \frac{\text{D.H.P.}}{\text{I.H.P.}} = \frac{102}{120} = 0.85 = 85$ per cent.

The relation between the actual and the theoretical efficiencies based upon the Rankine cycle, and upon the best recorded performances of modern engines, using saturated steam, is shown graphically in Fig. 144. (Also see Tables 39 and 40.) The theoretical curves are calculated upon the assumption of complete expansion, the back pressure being 14.7 pounds gauge for non-condensing and one pound absolute for condensing engines.

The highest recorded (1907) efficiency ratio for a steam engine using saturated steam is 83 per cent non-condensing and 76 per cent condensing (see Table 39), and 78.5 per cent for a condensing engine using superheated steam (Table 43).

160. Heat Losses in the Steam Engine. — The principal losses which tend to lower the efficiency of the steam engine are due to

(*a*) Presence of moisture in the steam at admission.
(*b*) Leakage past valves and piston.
(*c*) Cylinder condensation.
(*d*) Clearance volume.
(*e*) Incomplete expansion.
(*f*) Wire drawing.
(*g*) Friction.
(*h*) Radiation.

161. Moisture. — The presence of moisture in the steam pipe is due to condensation caused by radiation or to priming at the boiler. Unless removed by some separating device between boiler and engine the amount of moisture entering the cylinder may be from 1 to 5 per cent of the total weight of steam, and the work done per pound of fluid is correspondingly reduced. This loss should not be charged against the engine, however, and its performance should be reckoned on the dry steam basis. Experiments reported by Professor R. C. Carpenter (Trans. A.S.M.E., 15–438) in which water in varying quantities was introduced into the steam pipe, causing the quality of the steam to range from 99 per cent to 57 per cent, showed that the consumption

of *dry steam* per I.H.P. hour was practically constant, the water acting as an inert quantity. An efficient separator will remove practically all the entrained water.

Influence of Moisture on Steam Economy: Trans. A.S.M.E., 18-699; Benjamin, Heat and Steam; Perry, Steam Engine, p. 353; Rankine, Steam Engine, p. 407; Ripper, Steam Engine, p. 33; Ewing, The Steam Engine, p. 139.

162. Leakage of Steam. — The loss due to leakage is a variable factor depending upon the design and condition of the engine, and is greater with saturated than with superheated steam. The usual method of measuring leakage past the valves and piston while the engine is at rest is likely to give erroneous results as demonstrated by Callender and Nicolson (Peabody, "Thermodynamics," p. 351) in tests made on a high-speed automatic balanced valve engine and on a quadruple expansion engine with plain unbalanced slide valves. With the engines at rest they found that the leakage past valves and piston was insignificant, but when in operation the leakage from the steam chest into the exhaust was very considerable indeed. It was thought that a large proportion of the leakage was probably in the form of water formed by condensation of steam on the seat uncovered by the valve.

According to the report of the Steam Engine Research Committee (*Eng. Lond.*, March 24, 1905, p. 298), leakage through a plain slide valve is independent of the speed of the sliding surfaces, and directly proportional to the difference in pressure on the two sides; with well-fitted valves the leakage is never less than 4 per cent of the volume of steam entering the cylinders, and is often greater than 20 per cent.

Peabody, Thermodynamics, p. 350; Eng. Rec., May 22, 1897, p. 529; Barrus, Engine Tests, p. 251.

163. Cylinder Condensation. — A large percentage of the steam admitted to the cylinder is condensed, due to the absorption of heat by the relatively cool cylinder walls. Condensation continues during expansion until the temperature of the steam falls below that of the metal, when the process is reversed and a part of the moisture is re-evaporated. Unless the cylinder is one of a compound series, the heat absorbed from the cylinder walls during exhaust does no useful work and is lost. The condensation up to the point of cut-off (initial condensation) may amount to from 15 to 30 per cent, and is often as high as 50 per cent of the total weight admitted to the cylinder. The initial condensation becomes greater as the difference between initial and exhaust pressures is increased, and diminishes as the speed of the engine increases. Cylinder condensation and leakage are ordinarily

classified together, as there is no way of separating them accurately. They represent that part of the feed water which is not accounted for by the indicator diagram.

Tests of 20 simple high-speed engines by G. H. Barrus, Fig. 145, show some results obtained for various percentages of cut-off. Also see table compiled by C. H. Peabody, "Thermodynamics of the Steam

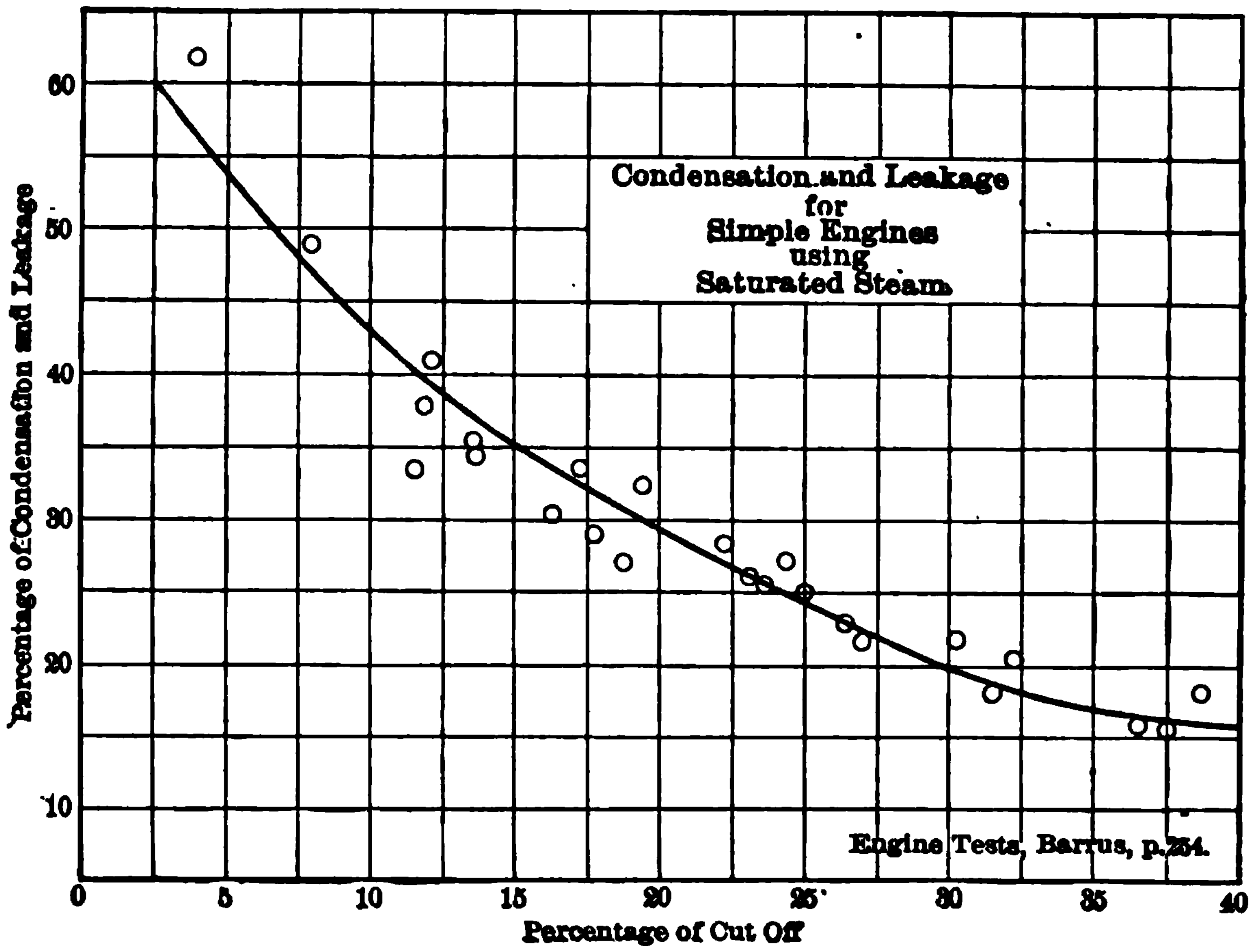


FIG. 145.

Engine," p. 336, showing analysis of the heat interchanges for a number of different types of steam engine.

The various heat losses, including cylinder condensation and leakage, are best determined by transferring the indicator diagram to the temperature entropy or $\theta\phi$ chart. (See Appendix C.) This is useful for certain scientific investigations, but is unnecessary for commercial tests.

Cylinder Condensation: Trans. A.S.M.E., I, 184, III, 215, IV, 88, VII, 375, XVIII, 950; Cotterill, Steam Engine, p. 331; Spangler, Steam Engineering, p. 228; Thurston, Manual of the Steam Engine, I, 271, 488, 585; Heck, Steam Engine, pp. 109, 113, 119; Ewing, Steam Engine, p. 148; Hutton, Heat Engines, p. 319; Peabody, Thermodynamics, pp. 241, 412; Reeve, Thermodynamics, p. 198; Wood, Thermodynamics, p. 212; Perry, Steam Engine, p. 78; Ripper, Steam Engine, p. 25.

Initial Condensation: Cotterill, Steam Engine, p. 274; Golding, $\theta\phi$ Diagram, p. 63; Marks, Steam Engine, p. 195; Peabody, Thermodynamics, p. 359; Popplewell, Heat Engine, pp. 323, 351; Reeve, Thermodynamics, pp. 156, 221; Ripper, Steam Engine, pp. 111, 168.

Condensation during Expansion: Trans. A.S.M.E., III, 286; Hutton, Heat Engines, pp. 223, 286; Pupin, Thermodynamics, p. 88; Rankine, Steam Engine, p. 385; Reeves, Thermodynamics, p. 221.

Entropy: Power, Jan. 21, 1908; Baynes, Thermodynamics, p. 94; Benjamin, Heat and Steam, p. 37; Berry, Temperature-Entropy Diagram; Boulvin, Entropy Diagram; Ewing, Steam Engine, p. 103; Golding, $\theta\phi$ Diagram; Hutton, Heat Engines, p. 276; Peabody, Thermodynamics, p. 97; Reeve, Thermodynamics, p. 39; Swinburne, Entropy; Wood, Thermodynamics, p. 136: Heck, Steam Engine, Chap. VI.

164. Clearance Volume. — The portion of the cylinder volume not swept through by the piston but which is nevertheless filled with steam when admission occurs is called the clearance volume. It is the space between the end of the piston when on dead center and the inside of the valves covering the ports. It varies from about 1 per cent of the piston displacement in very large engines with short steam passages to 10 per cent or more in small high-speed engines. When the steam retained in the clearance space is compressed to the initial pressure and expansion is carried down to the back pressure, the clearance has little effect upon the economy of the engine, but since expansion and compression are seldom complete in actual practice, the loss may be considerable. (Ripper, "Steam Engine," p. 103.) The shorter the cut-off the greater will be the ratio of the weight of cushion steam to that of the steam supplied and hence the greater the relative loss. In large slow-speed engines the loss may be insignificant if the clearance volumes are small, while in small high-speed engines it may be considerable.

The ratio of expansion is decreased by clearance; for example, an engine cutting off at one-fifth, neglecting clearance has an apparent ratio of expansion of 5, but if the clearance volume is 10 per cent the actual ratio is only 3.66. One of the few recorded tests relative to the influence of clearance on the economy of a high-speed engine was conducted on a 14 x 15 Allfree engine. (*Power*, May, 1901.) With a clearance volume of 2.2 per cent, initial pressure 105 pounds gauge, and 172 r.p.m., the best performance was 23.7 pounds of dry steam per I.H.P. hour. With the same steam pressure and speed, but with clearance volume increased to 6 per cent by the use of a shorter piston, the best performance was 28.3 pounds per I.H.P. hour. In both cases the compression was carried up to admission pressure.

Loss by Clearance in Steam Engines: Trans. A.S.M.E., XVIII, 176. *Clearance in Compound Engines:* ibid., I, 173. *Clearance in Multi-cylinder Engines:* ibid., XI, 151. *Effect of Clearance:* Ewing, Steam Engine, p. 145; Hutton, Heat Engines, p. 334;

Peabody, Thermodynamics, p. 407; Popplewell, Heat Engines, p. 332; Reeve, Thermodynamics, pp. 223, 256; Ripper, Steam Engine, pp. 63, 101; Spangler, Steam Engineering, p. 114; Wood, Thermodynamics, p. 197.

165. Loss Due to Incomplete Expansion and Compression. — Theoretically the loss due to incomplete expansion is considerable. For example, the theoretical steam consumption of a perfect engine (Rankine cycle) expanding from 120 pounds absolute to a condenser pressure of 2 pounds absolute is 9.6 pounds per horse-power hour. If the expansion were carried to only 5 pounds absolute, the exhaust pressure remaining the same, the steam consumption would be increased to 11.8 pounds per horse-power hour, a difference of 22 per cent for an increase in terminal pressure of only 3 pounds per square inch. The theoretical water rates for various terminal pressures are given below.

Terminal Pressure, Pounds per Square Inch Absolute.	Steam Consumption of Perfect Engine.	Terminal Pressure, Pounds per Square Inch Absolute.	Steam Consumption of Perfect Engine.
1	8.5	3	10.4
1.5	9.1	4	11.1
2	9.6	5	11.8
2.5	10	6	12.3

In actual engines expansion is seldom complete, since it would necessitate increased bulk and weight of engine, and the work done by the steam in the last stages would not compensate for the increased cost.

In single-cylinder engines maximum economy is effected when the terminal pressure is considerably above that of the exhaust, since the gain due to complete expansion is more than offset by the increased cylinder condensation. This is true to a certain extent in all engines irrespective of the number of cylinders. Tests by G.H. Barrus ("Engine Tests," 1900) to determine the terminal pressures effecting maximum economy for various types of engine gave results as follows:

	Terminal Pressure, Pounds Absolute.
Simple slide-valve engines, non-condensing	30 to 40
Simple slide-valve engines, condensing	25 to 30
Simple Corliss engines, non-condensing	20 to 25
Simple Corliss engines, condensing	15 to 18
Compound engines, non-condensing	18 to 22
Compound engines, condensing	3 to 5

In high-speed engines a certain amount of compression is desirable for its cushioning effect; outside of this mechanical feature compression may or may not be of benefit to the engine as will be explained later. Zuener in his treatise on theoretical thermodynamics proves deductively that in an engine with a large clearance volume the loss due to clearance is completely eliminated if the compression is carried up to admission pressure, a conclusion which tests by Jacobus, Carpenter, and others fail to confirm. A series of tests by Professor Jacobus (Trans. A.S.M.E., 15-918) on a 10 x 11 high-speed automatic engine at Stevens Institute show decreasing economy with increase of compression, the initial pressure, cut-off, and release remaining constant. The results were as follows:

Proportion of initial pressure up to which the steam is compressed	$\frac{5}{8}$	$\frac{3}{4}$	Full
Steam, pounds per I.H.P. hour	34.8	36.7	38

Tests by Carpenter (Trans. A.S.M.E., 16-957) on the high-pressure cylinders of the Corliss engine at Sibley College gave:

Compression, per cent	11.4	25	35.2
Brake horse power	30	29	26
Steam, pounds per B.H.P. hour	33	33.3	34

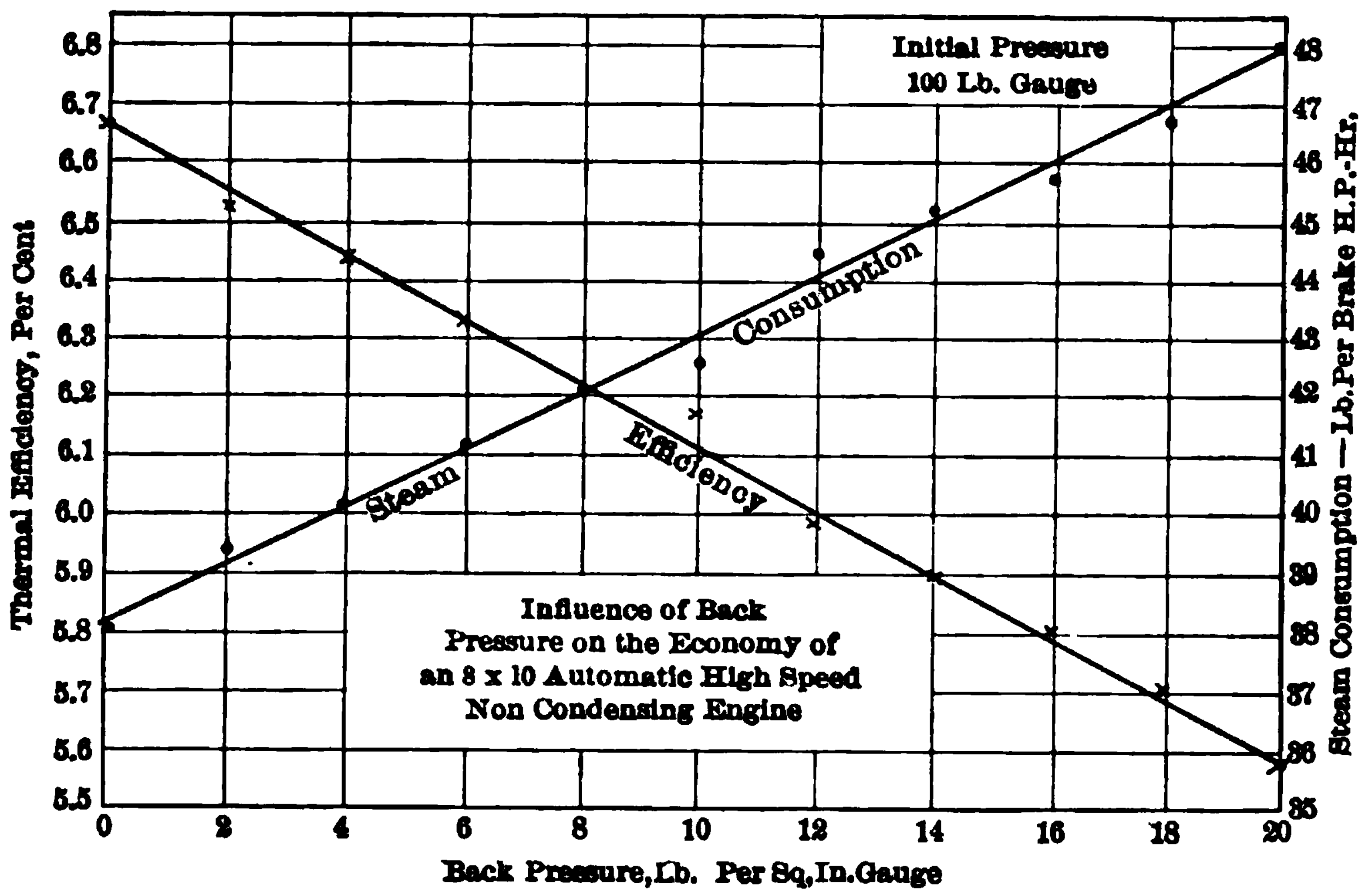


FIG. 145a.

Opposed to these figures are tests which show an improvement in economy when compression is increased.

Fig. 145a shows the influence of increasing back pressure on the

economy of an 8 x 10 automatic high-speed engine at Armour Institute of Technology.

Cut-off. — *Best for Different Pressures:* Trans. A.S.M.E., 4–89; *In Compound Engines:* ibid., p. 549; *Most Economical Point of:* ibid., 8–486; Hutton, Heat Engines, p. 232; Peabody, Thermodynamics, p. 210; Spangler, Steam Engineering, p. 109; Wood, Thermodynamics, pp. 200, 433; Klein, High-Speed Engines, p. 7; Ewing, Steam Engine, p. 84.

Ratio of Expansion: Trans. A.S.M.E., 2–19, 128, 10–576, 11–166; Ewing, Steam Engine, pp. 47, 159; Wood, Thermodynamics, pp. 154, 172, 197, 295; Rankine, Steam Engine, pp. 378, 553; Reeve, Thermodynamics, p. 228; Thurston, Manual of the Steam Engine, 1–271, 725, 2–14.

Incomplete Expansion: Boulvin, Entropy Diagram, p. 28; Cotterill, Steam Engine, p. 240; Perry, Steam Engine, p. 364; Popplewell, Heat Engines, p. 332; Peabody, Thermodynamics, p. 238; Reeve, Thermodynamics, p. 222; Heck, Steam Engine, p. 78.

Compression. — *Efficiency of Compression in Steam Engine:* Engr., Lond., Nov. 3, 1905, p. 434; *Compression as a Factor in Steam Engine Economy:* Trans. A.S.M.E., 14–1067; *Effect of Compression on Water Consumption:* ibid., 15–815; *Engine Compression:* ibid., 7–708; *In High-Speed Engines:* ibid., 7–202; *In Steam Cylinder:* ibid., 2–341.

Back Pressure. — *Back Pressure as Modifying Economy:* Trans. A.S.M.E., 18–283; *On Valves:* ibid., 3–150; *General:* Ewing, Steam Engine, pp. 84, 145; Klein, High-Speed Engines, p. 11; Perry, Steam Engine, p. 75; Reeve, Thermodynamics, p. 223; Ripper, Steam Engine, p. 53.

166. Loss due to Wire Drawing. — Wire drawing, or the drop in pressure due to the resistances of the ports and passages, has the effect of reducing the output of the engine to some extent, since the pressure within the cylinder is less than that at the throttle during admission and greater than discharge pressure at exhaust. The steam may be dried to a small extent during admission. In single-valve engines the effects of wire drawing are decidedly marked and the true points of cut-off and release are sometimes difficult to locate on the indicator card. In engines of the Corliss or gridiron-valve type the effects are hardly noticeable.

Wire Drawing: Trans. A.S.M.E., 2–344, 1–174; Ewing, Steam Engine, pp. 95, 143, 207; Boulvin, Entropy Diagram, p. 56; Popplewell, Heat Engine, p. 320; Rankine, Steam Engine, p. 413; Reeve, Thermodynamics, pp. 105, 221; Ripper, Steam Engine, p. 73; Wood, Thermodynamics, p. 195; Heck, Steam Engine, pp. 183, 224, 230.

167. Loss due to Friction of the Mechanism. — The difference between the indicated horse power and that actually developed is the power consumed in overcoming friction, and varies from 4 to 20 per cent of the indicated power, depending upon the type and condition of the engine. Engine friction may be divided into (1) initial or no-load

friction and (2) load friction. The stuffing-box and piston-ring friction is practically independent of the load, while that of the guides, bearings,

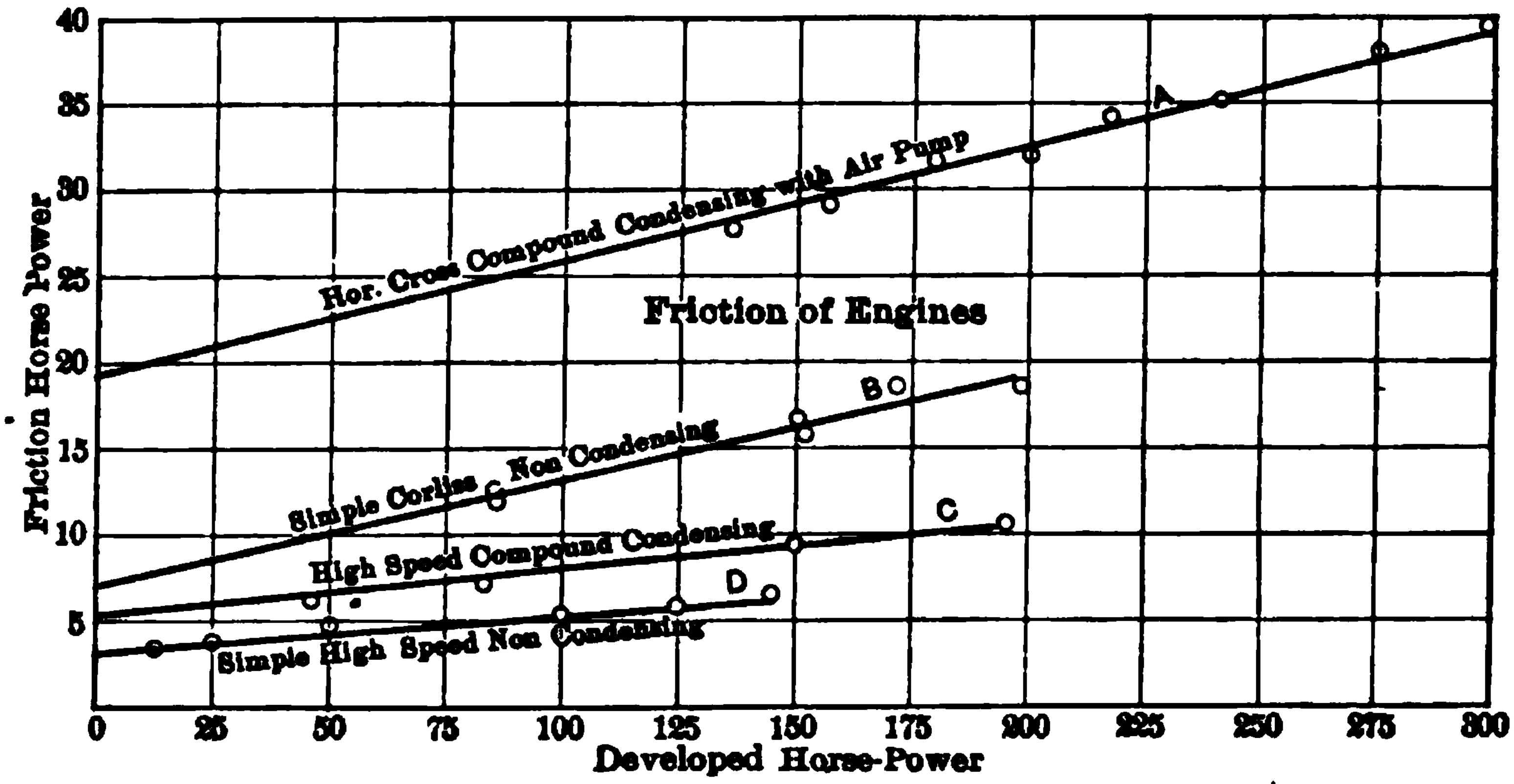


FIG. 146. Typical Curves of Steam Engine Friction.

and the like increases with the load. In Fig. 146, curve *A* gives the relation between the frictions for a four-slide-valve horizontal cross compound engine, and *B* that for a simple non-condensing Corliss.

TABLE 38.

DISTRIBUTION OF FRICTION IN SOME DIRECT-ACTING STEAM ENGINES.

(Thurston.)*

Parts of Engines where Friction is Measured.	Percentage of Total Engine Friction.				
	"Straight Line" Balanced Valve.	"Straight Line" Unbalanced Valve.	Traction Engine Locomotive Valve Gear.	Automatic Balanced Valve.	Condensing Engine Balanced Valve.
Main bearings.................	47.0	35.4	35.0	41.6	46.0
Piston and piston rod.........	32.9	25.0	21.0	49.1	21.8
Crank pin....................	6.8	5.1	13.0		
Crosshead and wrist pin.......	5.4	4.1			
Valve and valve rod..........	2.5	26.4	22.0	9.3	21.0
Eccentric strap..............	5.4	4.0			
Link and eccentric...........	....	....	9.0	....	....
Air pump....................	....	....	....	....	12.0
	100.0	100.0	100.0	100.0	100.0

* "Friction and Lost Work in Machinery," p. 13.

(Peabody's "Thermodynamics," pp. 433 and 437.) Curve C is plotted from the tests of a Reeves vertical cross compound condensing engine (*Engineering Record*, July 1, 1905, p. 24), and D from the test of an Ames simple high-speed non-condensing engine. (*Engineering Record*, Vol. 27, p. 225.) A large number of recorded tests show less friction at full load than at no load, but this is probably due to error or to variations in lubrication. With first-class lubrication it is usually sufficiently accurate to assume the friction to be constant and equal to the initial friction at zero load. The distribution of the frictional losses in a number of engines is given in Table 38.

Friction in Engines: Trans. A.S.M.E., 8–86, 10–10, 8–108, 9–74, 82, 7–639, 641, 1–153; Ewing, Steam Engine, p. 186; Ripper, Steam Engine, p. 275; Peabody, Thermodynamics, p. 430; Perry, Steam Engine, p. 270; Ripper, Steam Engine, 1–540; Heck, Steam Engine, 316–318.

The efficiency of the fluid in the steam engine cylinder may be increased by (1) raising the boiler pressure, (2) compounding, (3) use of reheater-receiver, (4) steam jacketing the cylinders, (5) increasing the rotative speed, (6) superheating, (7) diminishing the back pressure by securing a more perfect vacuum.

168. Effect of Increased Steam Pressure. — A consideration of the Rankine and Carnot cycles indicates that theoretically the greater the temperature range the greater will be the efficiency. (See Table 36.) In the actual engine the temperature range is most readily increased by raising the boiler pressure, since the limit of the back pressure is practically fixed by the cooling medium in the condenser. The theoretical gain resulting from increased pressure range is, however, very considerably affected by the increased losses due to cylinder condensation.

Fig. 147 shows the results of tests made at the Armour Institute of Technology on an 8 x 10 automatic high-speed piston-valve engine, showing marked gain with increase of initial pressure up to a certain point when the condensation losses became sufficiently great to neutralize the advantage which would otherwise be gained.

The following figures were obtained in tests of a small Willans engine, non-condensing, under different steam pressures:

Initial Pressure, Gauge.	Pounds Steam per I.H.P. Hour.	B.T.U. per I.H.P. per Minute.
36.3	42.8	700
51.0	36.0	595
74.0	32.6	544
85.0	29.7	495
97.0	26.9	450
110.0	27.8	465
122.0	26.0	436

Referring to Table 36, it may be noted that both the theoretical and the actual efficiencies increase very slowly for pressures above 150 pounds. Practically, gain in efficiency due to increasing the pressure

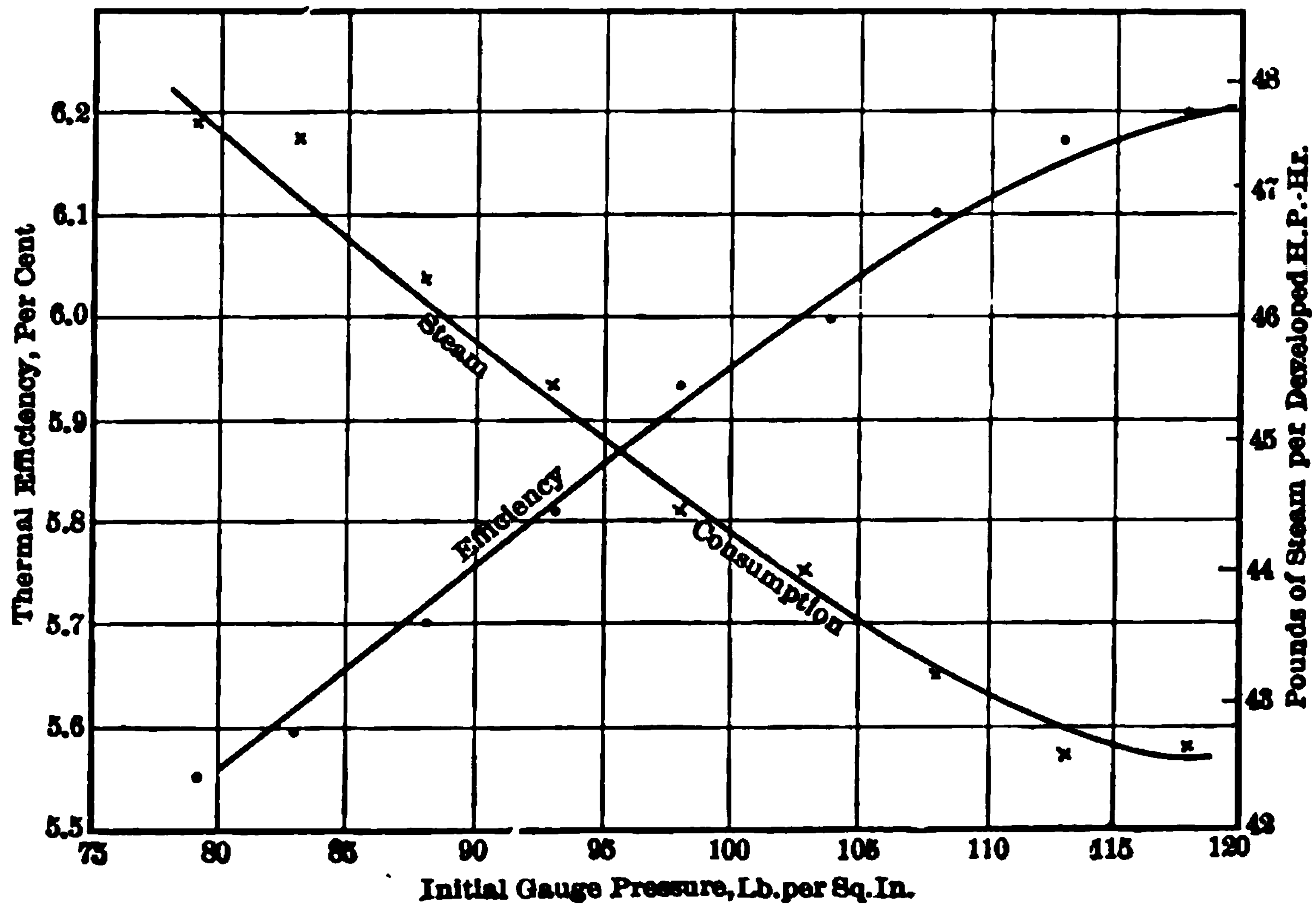


FIG. 147. Influence of Initial Pressure on the Economy of a Small, High-Speed, Non-Condensing Engine.

above about 200 pounds is at the expense of increased first cost and maintenance and is only resorted to when small weight and space are the most important considerations.

The range of pressures sanctioned by modern practice for different types of engines is as follows:

Type of Engine.	Range in Pressure (Gauge).	Average.
Simple slow speed	60–120	90
Simple high speed	70–125	100
Compound high speed, non-condensing	100–170	130
Compound high speed, condensing	100–160	125
Compound slow speed, condensing	125–200	150
Triple expansion, condensing	140–210	175
Quadruple expansion, condensing	125–225	200

Steam Pressure: Trans. A.S.M.E., 4–88, 5–269, 6–572; Peabody, Thermodynamics, p. 248; Ripper, Steam Engine, p. 306; Engine Tests, Barrus, p. 258.

169. Receiver-Reheaters. — The receivers between the cylinders of multi-expansion engines are frequently equipped with heating coils as illustrated in Fig. 332, the function of which is to superheat the exhaust steam before delivering it to the cylinder immediately follow-

ing, with a view of reducing the losses occasioned by cylinder condensation. The coils are supplied with live steam under boiler pressure and may serve to evaporate a portion of the moisture or to actually superheat the steam supplied to the following cylinder. The question of the propriety of using reheaters is an open one, since reliable data relative to their use are meager and discordant. The conditions under which the few recorded tests were made are too diverse to warrant definite conclusions. Some show an appreciable gain in economy, others a decided loss. A reheater is of little value in improving the thermodynamic action of the engine, and is probably a loss unless it produces a superheat of at least 30 degrees F., and to be fully effective should superheat above 100 degrees F. (L. S. Marks, Trans. A.S.M.E., 25–500.) The effectiveness of the reheater will evidently be increased by the removal of the greater portion of the moisture from the exhaust steam before it enters the receiver. In the 5000-horse-power engine at the Waterside Station in New York it was shown that both jackets and reheaters, either together or alone, were practically valueless throughout the working range of load. (*Power*, July, 1904, p. 424.) Many similar cases may be cited which show no gain in economy with the use of the reheaters. On the other hand, with properly proportioned reheaters, the gain may be considerable and particularly with superheated steam. Practically all European engines operating with highly superheated steam are equipped with receiver-reheaters. In all cases the reheater effects a great reduction in the condensation in the low-pressure cylinders, but the resulting gain, considering the condensation in the reheater coils, may be little if any.

In triple expansion pumping engines receiver-reheaters are found to effect an appreciable gain in economy, and practically all such engines are equipped with them. In electric traction work or where the load is a widely fluctuating one the reheater has been virtually abandoned. Apart from the consideration of fuel economy, all tests show a marked increase in the indicated power of the low-pressure cylinder (5 to 15 per cent), and to that extent it increases the capacity of the entire engine. (G. H. Barrus, *Power*, September, 1903, p. 516.)

Receivers: Trans. A.S.M.E., 1-174, 178, 9-549; Ewing, Steam Engine, p. 222; Holmes, Steam Engine, p. 456; Spangler, Steam Engineering, p. 249; Whitham, Steam Engine, p. 395; Power, June, 1896, p. 20; ibid., Nov., 1905, p. 684.

Receiver-Reheaters: Trans. A.S.M.E., 1-178, 17-509, 25-443; Power, Sept., 1903, p. 516; Am. Elecn., Oct., 1902, p. 480; ibid., July, 1904, p. 328; Eng. Rec., May 9, 1903, p. 496; Engineering, Aug. 8, 1902, p. 197; ibid., Aug. 21, 1902, p. 125; Eng. Rec., May 28, 1904, p. 690.

170. Jackets. — If the walls of the cylinder are made double and the space between is filled with live steam under boiler pressure, the

cylinder is said to be steam jacketed. The function of the jacket is to reduce initial condensation by maintaining the temperature of the internal walls as nearly as possible equal to that of the entering steam. The heat given up by the jacket steam, and the resulting condensation, is usually a smaller loss than would otherwise result from cylinder condensation. However, tests of numerous engines with and without steam jackets do not agree as to the conditions under which their use is profitable, the apparent gain ranging from zero to 30 per cent. According to Peabody, a saving of 5 to 10 per cent may be made by jacketing simple and compound condensing engines, and a saving of 10 to 15 per cent by jacketing triple expansion engines of 300 horse power and under. On large engines of 1000 horse power or more the gain, if any, is very small. (Peabody, "Thermodynamics," p. 400.)

Other things being equal, the smaller the cylinder and the lower the piston speed the greater is the value of the jacket. Experiments show no advantage in increasing the jacket pressure more than a few pounds above that of the initial steam in the cylinder, and it is usual to reduce the pressure in the jackets of the second and succeeding cylinders of multi-expansion engines. (Ripper, "Steam Engine," p. 170.)

To be effective, jackets should be well drained, kept full of live steam, and the water of condensation returned directly to the boiler.

Pumping engines and other slow-speed engines running at practically constant load are generally jacketed, but in street-railway work and in the majority of manufacturing plants carrying fluctuating load, jackets are not considered advantageous.

Whatever may be the actual economy due to jacketing, there is no question but that the jacket greatly influences the action of the steam in the cylinders, and whether beneficially or not depends upon the design and construction of the engine. Unless otherwise specified, manufacturers usually build their engines without jackets.

Steam Jackets: Trans. A.S.M.E., 1-175, 190, 2-198, 9-554, 11-141, 149, 328, 1038, 12-873, 462, 13-176, 14-1356, 15-779, 137; Golding, $\theta\phi$ Diagram, p. 39; Hutton, Heat Engines; Perry, Steam Engine, p. 369; Rankine, Steam Engine, p. 395; Reeve, Thermodynamics, p. 29; Ripper, Steam Engine, p. 166; Thurston, Manual of the Steam Engine, 1-598, 622; Peabody, Thermodynamics, p. 322; Heck, Steam Engine, p. 123; Am. Mach., Jan. 30, 1896, p. 126; Engr., Lond., April 21, 1905, p. 401; Engineering, Jan. 30, 1905, p. 829; Eng. Rec., April 16, 1898, p. 423; Power, Feb., 1898, p. 17; ibid., Feb., 1899, p. 9; Eng. Mag., June, 1898, p. 479, June, 1899, p. 496, Aug., 1905, p. 755.

Increasing Rotative Speed: See High-Speed *vs.* Low-Speed Engines, paragraph 172.

Compounding: See Compound Engines, paragraph 177.

Reducing Back Pressure: See Influence of Condensing, paragraph 179.

Superheating: See paragraph 181.

171. Single- and Double-Acting Engines. — When steam pressure is exerted on only one end of the piston the engine is said to be single acting, and when exerted alternately on one side and the other is said to be double acting. For high speed, minimum wear and tear, and comparatively cheap construction the single-acting engine offers some advantages. The Westinghouse Standard and the Willans Central-Valve engines are typical of this class. Silent running at high speed is possible because the pressure on the crank pin is not reversed. The output is only half that of a double-acting engine of the same size and speed, but a much higher rotative speed is permissible, which somewhat offsets this disadvantage. Single-acting engines have been operated successfully with speed as high as 1000 r.p.m., while double-acting engines seldom exceed 350 r.p.m. and that only for strokes less than 12 inches.

Single-Acting Engines. — *Comparison between Different Types of Engines:* Trans. A.S.M.E., 2–294. *Economy of Single-Acting Expansion Engines:* ibid., 3–252. *Single-Acting Compound:* ibid., 12–275. *Steam Distribution in Compound:* ibid., 13–557; Ewing, Steam Engine, p. 371; Rankine, Steam Engine, p. 478; Hutton, Power Plants, p. 77.

Double-Acting Engines: Hutton, Power Plants, p. 73; Rankine, Steam Engine, p. 50; Ewing, Steam Engine, p. 20; Le Van, Steam Engine, p. 240.

172. High- and Low-Speed Engines. — High rotative speed does not necessarily mean high piston speed. An 8 x 10 engine running at 300 r.p.m. has a piston speed of only 500 feet per minute, whereas a 36 x 72 Corliss running at 60 r.p.m. has a piston speed of 720 feet per minute. The classification " high speed " and " low speed " refers to rotative speed only, the former above and the latter below say 150 r.p.m.

On account of the reduction of thermodynamic wastes, a high-speed engine should give theoretically a higher efficiency than the same engine at a lower speed, all other conditions being the same. The effect of speed upon economy is decidedly marked in engines and pumps taking steam full stroke. For example, tests of a 12 x $7\frac{1}{2}$ x 12 simplex direct-acting steam pump at Armour Institute of Technology showed a steam consumption of 300 pounds per I.H.P. hour at 10 strokes per minute, and only 99 pounds at 100 strokes per minute. (See Figs. 274 and 275.)

Tests of engines using steam expansively, however, do not furnish conclusive evidence on this point, some showing a decided gain (Peabody, "Thermodynamics," p. 425), others little or no gain (Barrus, "Engine Tests," p. 260). For example, a small Willans engine showed an increase in economy of 20 per cent in increasing the rotative speed

from 111 to 408 r.p.m. (Peabody, "Thermodynamics," p. 402), whereas the compound locomotives at the Louisiana Purchase Exposition showed a loss in economy for the higher speeds (Publication by the Pennsylvania Railroad Company). On the other hand, a comparison of the performances of high- and low-speed Corliss engines shows little difference in economy, and a general comparison between high- and low-speed engines furnishes little information, since nearly all high-speed engines are of a different class from the low-speed ones. High-speed engines are comparatively small in size, require larger clearance volume, and are usually fitted with a single valve. Rotative speed is limited by design, material, workmanship, and cost of subsequent maintenance. Speeds of 400 r.p.m. and more are not unusual with single-acting engines, whereas 300 r.p.m. is about the limit for double-acting machines with strokes over 12 inches in length. A comparison of tests of high-speed and low-speed engines in this country, irrespective of design and construction, shows the former to be less economical than the latter in most cases. In Europe high-speed engines are developed to a high degree of efficiency, and their performances are comparable with the best grade of low-speed engines.

High-speed engines as a class have the advantage of being more compact for a given power, are simple in construction and relatively low in first cost; on the other hand, they are subject to comparatively rapid depreciation, excessive vibration, and are less economical in steam consumption.

High- and Low-Speed Engines.—Effect of Speed on Condensation: Peabody, Thermodynamics, p. 424. *Effect of Speed on Economy:* Barrus, Engine Tests, p. 257; Trans. A.S.M.E., 7–397, 2–198. *Limitation of Speed:* Trans. A.S.M.E., 14–806. *Effect of Speed on Economy:* Ripper, Steam Engine, p. 317.

173–4. High-Speed Single-Valve Simple Engines.—This style of engine is made in sizes varying from 10 to 500 horse power. The cylinder dimensions vary from 4 x 5 to 24 x 24 and the rotative speed from 300 to 175 r.p.m.

When ground is limited or costly and exhaust steam is necessary for heating or manufacturing purposes, the high-speed non-condensing engine is most suitable for horse powers of 200 or less, being compact, simple in construction and operation, and low in first cost. For sizes larger than this the compound engine would probably prove a better investment, except in cases where fuel is very cheap or large quantities of exhaust steam are to be used for manufacturing purposes.

Small high-speed engines are seldom operated condensing, since the

gain due to reduction of back pressure is more than offset by the extra cost of the condenser and appurtenances.

Engines are ordinarily rated at about 75 per cent of their maximum output. For example, a 12 x 12 non-condensing engine running at 300 r.p.m., with initial steam pressure of 80 pounds gauge, is normally rated at 70 horse power, though it is capable of developing 90 horse power at the same speed.

The steam consumption of high-speed single-valve non-condensing engines at full load ranges from 27 to 50 pounds per indicated horse-power hour, depending upon the size of the unit and the conditions of operation. An average for good practice is not far from 30 pounds. With superheated steam a steam consumption as low as 18 pounds per horse-power hour has been recorded.

Table 39 gives the steam consumption of a number of single-valve high-speed engines running condensing and non-condensing, and

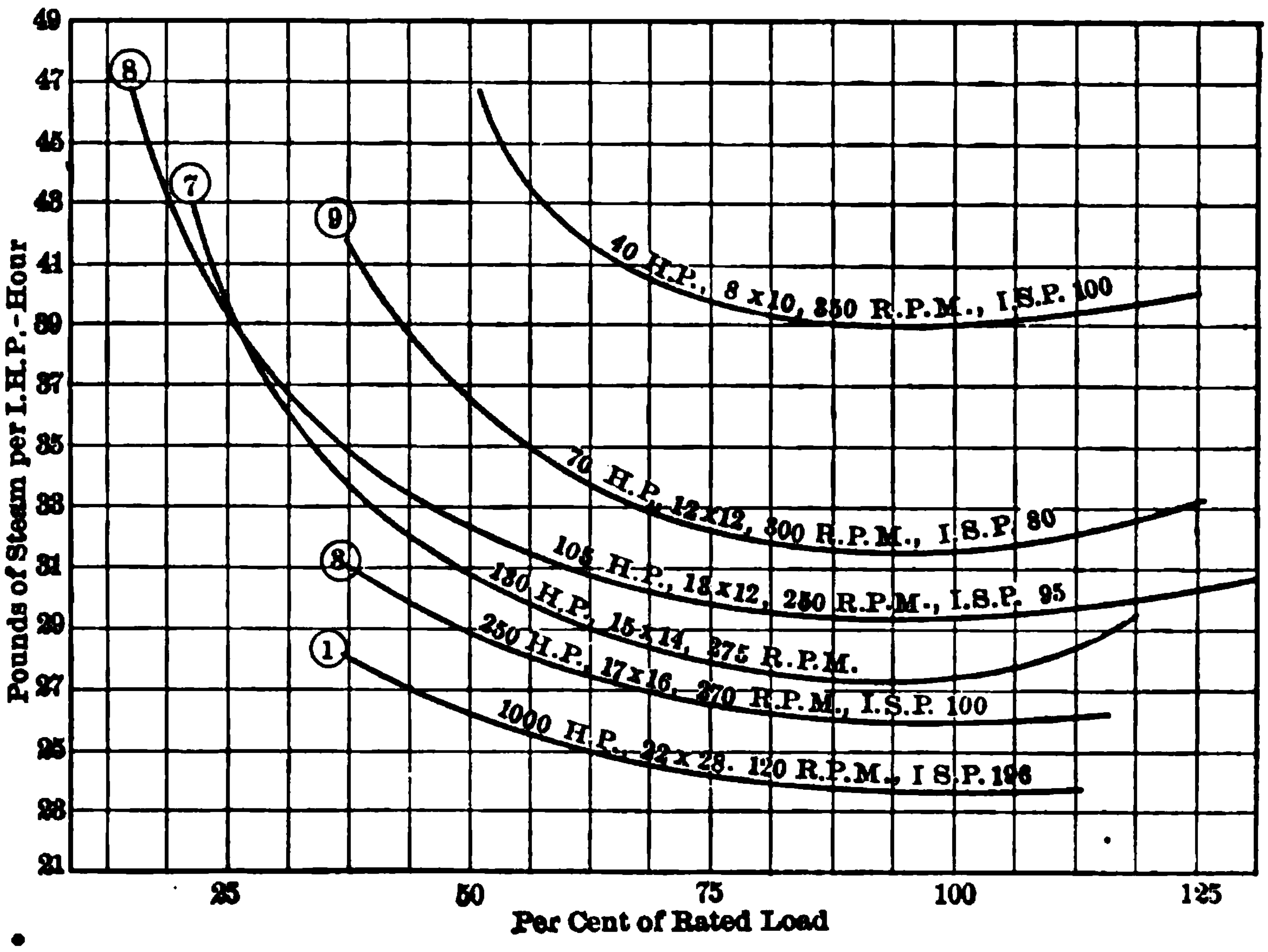


FIG. 148. Typical Economy Curves of High-Speed, Single-Valve, Non-Condensing Engines. Saturated Steam.

Fig. 148 shows some of the results for different loads. The steam consumption is fairly constant from 50 per cent of the rated load to 25 per cent overload, but for earlier loads the economy drops off rapidly. The desirability of operating the engine near its rated load is at once apparent. The curves show a marked economy in favor of the larger cylinders, but the engines are not of the same make, and the conditions of operation are somewhat different.

FIG. 148a. Assembly of Valve Gear; Typical Corliss Engine.

FIG. 148b. Section Through Cylinder; Typical Corliss Engine.

Fig. 148c. Assembly of Governor and Link Mechanism; Typical Corliss Engine

The most economical cut-off for a simple engine is about one-third to one-fourth stroke when running non-condensing, and about one-sixth when condensing.

An excellent performance for a small high-speed single-valve simple engine using saturated steam is shown by the Reeves piston valve engine (No. 6, Table 39) in a test made by Professor R. C. Carpenter. When running non-condensing with initial steam pressure of 114 pounds gauge the lowest steam consumption was 28 pounds per I.H.P. hour and 31.8 pounds per B.H.P. hour. Referred to the heat-unit basis, this gives 470 B.T.U. per I.H.P. per minute, non-condensing, and 450 B.T.U. per I.H.P. per minute, condensing.

A test of a 12 x 12 simple slide-valve Buffalo engine (No. 10, Table 39) by Professor Reeves gave 30.6 pounds of steam per I.H.P. hour, initial gauge pressure 79.3 pounds. As far as the *weight of steam* and B.T.U. per I.H.P. per minute are concerned, the Reeves engine shows the better economy, but referred to the performance of the ideal engine the Buffalo engine is the *more nearly perfect* of the two. The latter consumed 510 B.T.U. per I.H.P. per minute, and a perfect engine working through the same range in temperature would require 324 B.T.U. per I.H.P. per minute; hence the efficiency ratio or the degree of perfection of the latter is $324 \div 510 = 63.5$ per cent as against 57.5 per cent for the former.

Locomotive No. 1499 of the Pennsylvania system holds the record for economy in steam consumption for a single-valve non-condensing engine (No. 9, Table 39). The steam consumption is 23.4 pounds per I.H.P. hour (initial gauge pressure 196 pounds per square inch), which corresponds to a heat consumption of 398 B.T.U. per I.H.P. per minute. So far as the writer knows, this is the best recorded performance for a single-valve non-condensing engine using saturated steam.

The performance of the Ames engine (No. 7, Table 39) is one of the best recorded for so low an initial pressure.

A small single-acting Willans engine (No. 1, Table 39) holds an exceptional record for economy for a very small single-valve high-speed engine, having given a steam consumption of 26 pounds per I.H.P. hour, non condensing, initial gauge pressure 122 pounds, corresponding to a heat consumption of 436 B.T.U. per I.H.P. per minute. All of these performances are at the best rating of the engines. For a continually changing load, as in electric-lighting service, the average steam consumption is considerably greater than that at full load and depends upon the " load factor " (the ratio of the actual to the rated load). This is clearly shown in the curves of the steam consumption, Figs. 148 and 143.

TABLE 39. — EXAMPLES OF STEAM-ENGINE ECONOMY.

SIMPLE ENGINES, SATURATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions.	I.H.P.	Initial Pressure, Lbs. Gauge.	Back Pressure, Lbs. per Sq. In., Abs.	M.E.P.	R.P.M.	Lbs. of Dry Steam per I.H.P. Hour.	B.T.U. per I.H.P. per Minute.	B.T.U. per I.H.P. per Minute, Perfect Engine.	Efficiency Ratio, per Cent.	Mechanical Efficiency, per Cent.
					SINGLE-VALVE, NON-CONDENSING.								
1	Willans	Peabody, Thermodynamics	14 × 6	33.6	122.0	Atmo.	....	400	26.0	436	263	60.4	....
2	Willans	Peabody, Thermodynamics	14 × 6	16.5	36.3	..do..	....	393	42.8	704	570	81.0	....
3	Locomotive, Purdue	Trans. A.S.M.E., Vol. 14, p. 826	17 ×14	399.0	110.0	..do..	54.0	136	24.97	420	273	65.0	....
4	Westinghouse Standard	Shop test	20 ×16	257.0	100.0	..do..	36.0	275	26.19	439	286	65.2	....
5	Buffalo forge	Elec. World, Sept. 1, 1904, p. 407	12 ×12	121.0	124.0	..do..	58.5	302	27.5	462	265	57.5	....
6	Reeves	Elec. World, Oct. 1, 1904, p. 587	15 ×14	120.0	114.0	..do..	35.0	275	28.0	470	270	57.5	92.5
7	Ames	Eng. Record, July 6, 1901, p. 7	13 ×12	105.0	95.0	..do..	52.0	250	29.1	493	293	59.4	....
8	Ames	Meyer, Steam Power Plants, p. 56	17 ×16	248.0	100.0	..do..	50.0	270	26.0	435	286	65.7	....
9	Locomotive, No. 1499, Penn. System.	Locomotive Tests, 1904, at Louisiana Exposition.	22 ×28	975.0	196.0	..do..	75.6	120	23.4	398	222	55.9	91.0
10	Buffalo forge	Elec. World, Sept., 1904, p. 407	12 ×12	74.0	79.3	..do..	35.6	304	30.6	510	324	63.5	9.57
					SINGLE-VALVE, CONDENSING.								
11	Willans	Peabody, Thermodynamics	14 × 6	33.2	70.0	1.0	....	383.0	22.2	410	169	41.3	....
12	Reeves	Elec. World, Oct. 1, 1904, p. 587	15 ×14	140.0	114.0	3.2	41.0	275.0	26.0	466	186	39.7	95.8
13	Buffalo forge	Elec. World, Sept. 10, 1904, p. 407	12 ×12	86.0	80.0	3.0	40.5	310.0	27.5	490	197	40.0	95.0
14	Piston valve	Barrus, Engine Tests, p. 95	18½ ×30	204.6	69.3	2.6	38.1	29.3	27.15	435	199	41.0	....
					FOUR-VALVE, NON-CONDENSING.								
15	Corliss, jacketed	Peabody, Thermodynamics	21.6 ×43.3	237.0	103.5	Atmo.	42.1	62.7	21.5	358	280	78.0	....
16	Fleming	Prof. Carpenter, June 28, 1905, at Cornell University.	19 ×19.0	217.9	120.5	..do..	39.0	205.9	22.46	378	265	70.0	....
17	Fleming	Prof. Spangler, June 6, 1905, at University of Pennsylvania.	16 ×16.0	132.0	125.4	..do..	39.1	210.0	22.24	374	260	69.5	95.0
18	Corliss	Peabody, Thermodynamics	18 ×48.0	120.0	96.0	..do..	25.7	76.0	23.9	400	292	73.5	....
19	Corliss	Barrus, Engine Tests, p. 47	28½ ×59½	506.0	101.0	..do..	41.0	64.8	25.8	432	286	66.2	....
20	Corliss	Barrus, Engine Tests, p. 126	16 ×42.0	342.0	91.6	..do..	48.4	84.9	25.9	434	312	72.0	....
					FOUR-VALVE, CONDENSING.								
21	Corliss, jacketed	Peabody, Thermodynamics	21.6 ×43.3	155.0	103.8	1.2	32.0	60.0	16.5	302	161	53.3	....
22	Poppet valves, jacketed	Zeit. d. V. D. Ing., Aug., 1905, p. 1310.	22.6 ×45.0	262.0	79.0	1.36	30.2	47.6	15.0	275	175	63.8	....
23	Gridiron valves	Barrus, Engine Tests, p. 101	34½ ×60.0	613.0	82.3	1.0	37.2	60.0	18.5	342	164	48.0	....
24	Corliss	Barrus, Engine Tests, p. 118	32 ×60.0	554.0	67.5	2.9	38.2	59.1	19.45	342	206	60.2	....
25	Slide valves	Barrus, Engine Tests, p. 88	18 ×30.0	213.0	67.0	2.2	33.7	165.0	22.0	397	195	49.1	....
26	Corliss	Peabody, Thermodynamics	18 ×48.0	145.0	96.0	1.5	30.9	76.0	19.4	356	171	48.0	....

The performances given in Table 39 are exceptional. It is not advisable to count on a better steam consumption for this type of engine than 30 to 35 pounds of steam per I.H.P. hour.

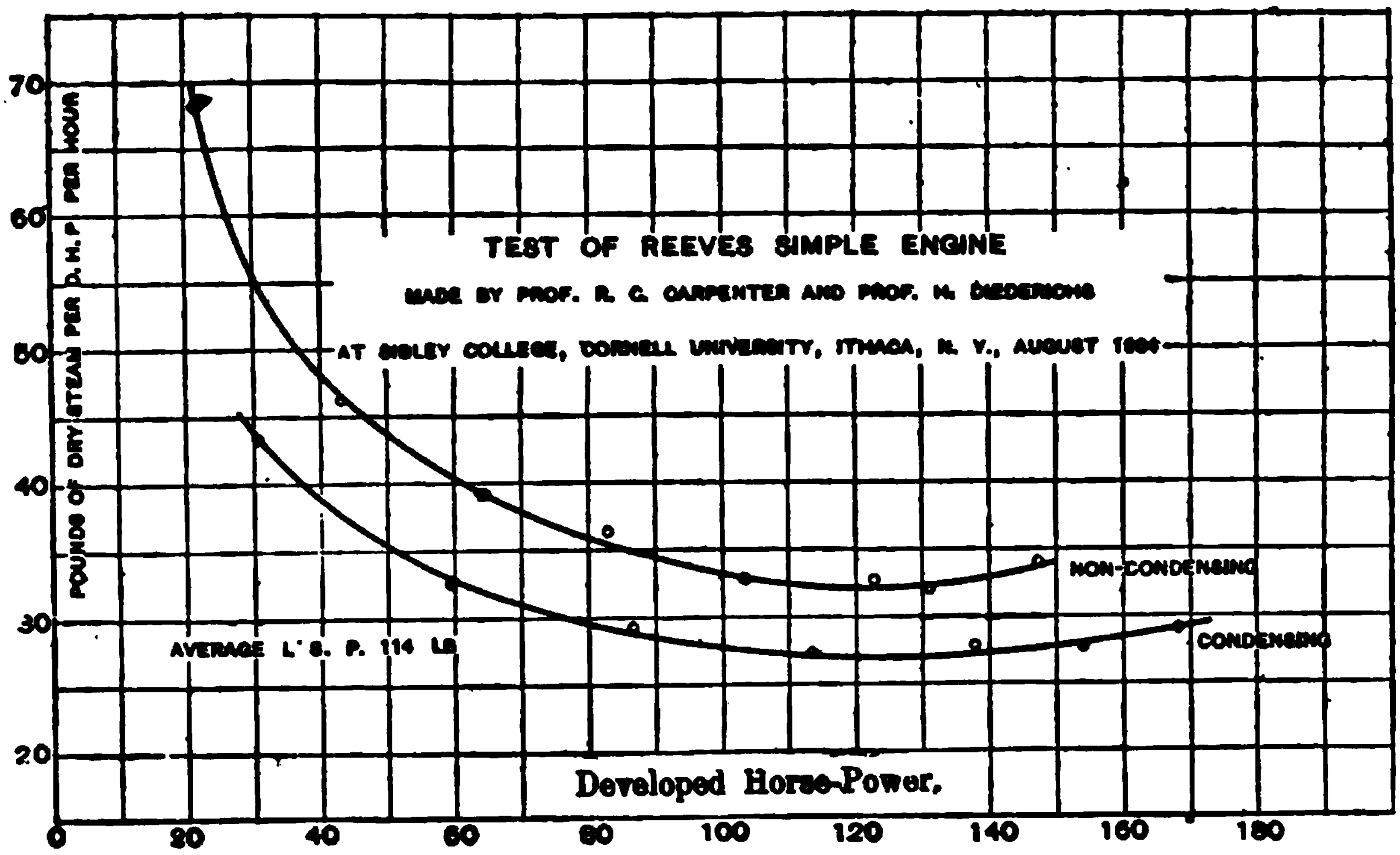


FIG. 149.

Fig. 149 shows the effects of condensing on a typical single-valve high-speed engine. The gain in fuel economy may be only an apparent one, since the steam consumption of the condensing apparatus should be rightfully charged to the engine.

When used in connection with heating plants or manufacturing plants requiring large quantities of exhaust steam the thermal efficiency is very high and may reach 85 per cent as against 22 per cent for the best compound condensing engine. In general when the requirements for exhaust steam are in excess of the steam consumption of a simple non-condensing engine a high-grade economical engine is without purpose.

175. High-Speed Multi-Valve Engines. — The steam distribution in a single-valve engine may give good economy for a very small range in load but be far from satisfactory for a wide range. This must necessarily be so since admission, cut-off, release, and compression are all functions of one valve, and any change in one results in a change of the others. To obviate the limitations of the single valve, many builders design engines with two or more valves. With a two-valve engine cut-off is independent of the other events, and with four valves all events are independently adjustable. In addition to the flexibility of the valve gear, the chief feature of the four-valve engines lies in the reduction of clearance volume which is made possible by placing the valves directly

over the ports. The valves may be of the common slide-valve or rotary type. As a class, four-valve engines are more economical than those having a less number of valves. The advantages and disadvantages of the four-valve over the single-valve engines may be tabulated as below.

ADVANTAGES.	DISADVANTAGES.
1. Better steam distribution.	1. Increased number of parts.
2. Better regulation.	2. Increased first cost.
3. Reduced clearance volume.	3. Requires greater attention.
4. Less valve leakage.	
5. Better economy.	

The steam consumption of a high-speed four-valve non-condensing engine varies from 22 to 35 pounds of saturated steam per horse-power hour, with an average not far from 27 pounds. With superheated steam the steam consumption may run as low as 18 pounds per horse-power hour.

An exceptional performance for a simple unjacketed high-speed four-valve engine is that of Engine No. 17, Table 39. With initial gauge pressure of 125 pounds the steam consumption is 22.24 pounds per I.H.P. hour, corresponding to a heat consumption of 374 B.T.U. per I.H.P. per minute.

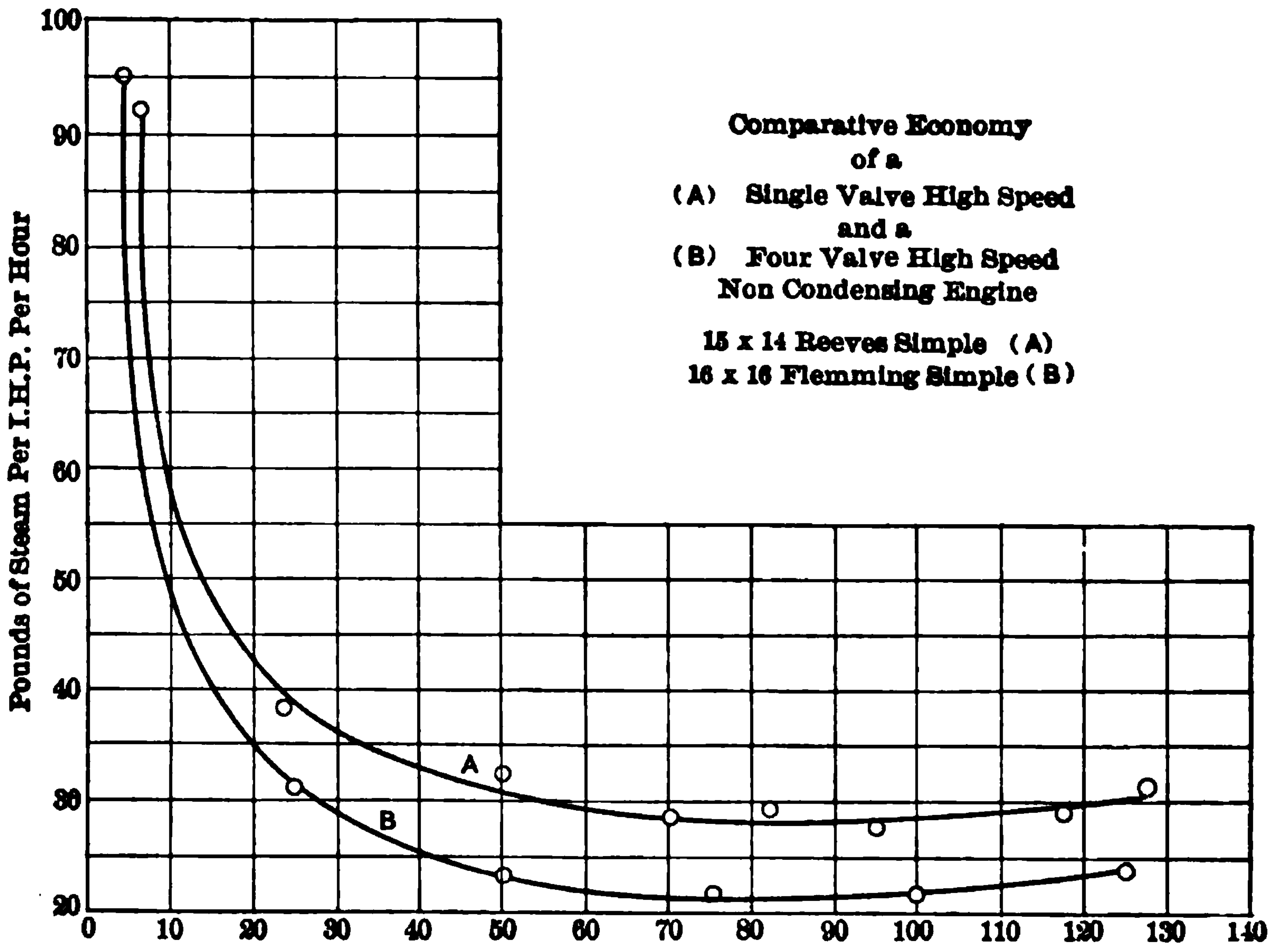


FIG. 150.

Fig. 150 gives a comparison between a single-valve and a four-valve high-speed engine, and though the engines differ slightly in size, the

conditions of operation were comparable and the marked gain in economy of the latter over the former is apparent. Both performances are exceptional, and a 10 to 15 per cent greater steam consumption may be expected in average good practice.

As a general rule single-valve simple engines do not exceed 500 horse power in size for stationary work, whereas 1000 horse power is not an uncommon size for the multi-valve type.

High-Speed Engines, General Description: Trans. A.S.M.E., 2–75; ibid., 17–117; Engr. U.S., Jan. 15, 1903; Engr., Lond., April 15, 1904, p. 379, April 29, p. 433, May 13, p. 478, May 20, p. 529. *Proportions of High-Speed Engines:* Trans. A.S.M.E., 8–191; Klein, High-Speed Engines; Hutton, Power Plants, p. 70, Thurston, Stationary Steam Engines.

Tests of Simple High-Speed Engines: Am. Elecn., April, 1901, p. 197, Dec., 1903. p. 581; Trans. A.S.M.E., 11–723, 18–795; Elec. World, May 20, 1903, p. 897, Sept. 10, 1904, p. 404, Oct. 1, 1904, p. 587, Feb. 17, 1906, p. 369; Engr. U.S., June 1, 1903, p. 416, Nov. 1, 1904, p. 758; Engineering, July 22, 1898, p. 116; Eng. News, Dec. 3, 1903, p. 493; Eng. Rec., July 6, 1901, p. 225; Machinery, May, 1903, p. 481; Power, Jan., 1904, p. 44, Nov., 1904, p. 651, Jan., 1905, p. 56; Stevens Indicator, Jan., 1900, p. 9; St. Ry. Jour., Oct., 1904, p. 673; Technology Quarterly, Sept., 1899, p. 255.

176. Medium and Low-Speed Multi-Valve Engines. — A comparison of tests of high and low-speed single-valve engines irrespective of design and construction shows the former as a class to be less economical than the latter. With four-valve engines there is no such disparity, and the high-speed type has shown just as good economy as the slow-speed class. For example, Engine No. 17, Table 39, with Corliss valves and a speed of 210 r.p.m., gives practically the same economy as Corliss engine No. 15 operating at 62 r.p.m. By far the greater number of simple multi-valve slow-speed simple engines are of the Corliss type. They range in size from 50 to 3000 horse power, with cylinders varying from 12 x 30 to 48 x 72. The smaller sizes with trip-valve gear run at 90 to 100 r.p.m., and the larger at 50 to 75 r.p.m. Without the trip gear, speeds of 150 r.p.m. are not uncommon, but at this speed they are usually classified as high-speed engines.

Table 39 gives the steam consumption, condensing and non-condensing, of a number of four-valve slow-speed simple engines.

Engine No. 15 shows an unusual performance for a simple Corliss engine operating both condensing and non-condensing. With initial gauge pressure of 103.5 pounds, the minimum steam consumption is 21.5 pounds per I.H.P. hour for the non-condensing run and 16.5 pounds for the condensing run. This corresponds to 358 B.T.U. per I.H.P. per minute, non-condensing, and 302 B.T.U. per I.H.P. per minute,

condensing. The efficiency ratios are 78.0 per cent and 53.2 per cent respectively. The cylinder was jacketed.

Attention is also called to the record of engine No. 22, Table 39, which is of the Sulzer type with four balanced poppet valves, heads and cylinder barrel jacketed. With 79 pounds initial pressure and a vacuum of 1.36 pounds absolute, the steam consumption is 15 pounds per I.H.P. hour, corresponding to a heat consumption of 275 B.T.U. per I.H.P. per minute.

177. Compound Engines. — Compound engines may be divided into three classes, tandem, cross compound, and duplex. In the tandem the two cylinders are end to end, in the cross compound side by side, and in the duplex one above the other. The tandem and duplex compounds have the advantage of (1) compactness for a given power, (2) less complication and fewer parts, and (3) low first cost. The crank effort is more variable than in the cross compound. In very large engines the low-pressure stage is generally divided between two cylinders of equivalent size to avoid an excessively large single cylinder and to distribute the crank effort. High-speed non-condensing compounds are ordinarily of the tandem type and are finding much favor in isolated station work, as in the power plants of tall office buildings where ground space is limited, though the duplex compound is sometimes used. The vertical or horizontal cross compound is generally installed in street-railway plants.

Cylinder ratios for high-speed single-valve compound engines vary from about 1 to 2½ with 100 pounds pressure to about 1 to 3 with a pressure of 150 pounds, and for slow-speed condensing engines from 1 to 3 with 125 pounds pressure to about 1 to 4 with a pressure of 175 pounds. G. I. Rockwood recommends a ratio as high as 7 to 1, and a number of engines designed along this line have shown exceptional economy. A cross compound Corliss engine at the Atlantic Mills, Providence, R.I., with cylinders 16 and 40 x 48 (ratio 6.128 to 1) gave the low steam consumption of 11.2 pounds of steam per I.H.P. hour, corresponding to a heat consumption of 222 B.T.U. per I.H.P. per minute. The 5500-horse-power engines of the New York Edison Company have a cylinder ratio of 6 to 1. The great majority of compound engines, however, have cylinder ratios of 4 to 1 or less. The 8000-horse-power engines of the Interborough Rapid Transit system have a ratio of 4 to 1, and the 4000-horse-power units of the Metropolitan Elevated Company, New York, a ratio of 3.5 to 1.

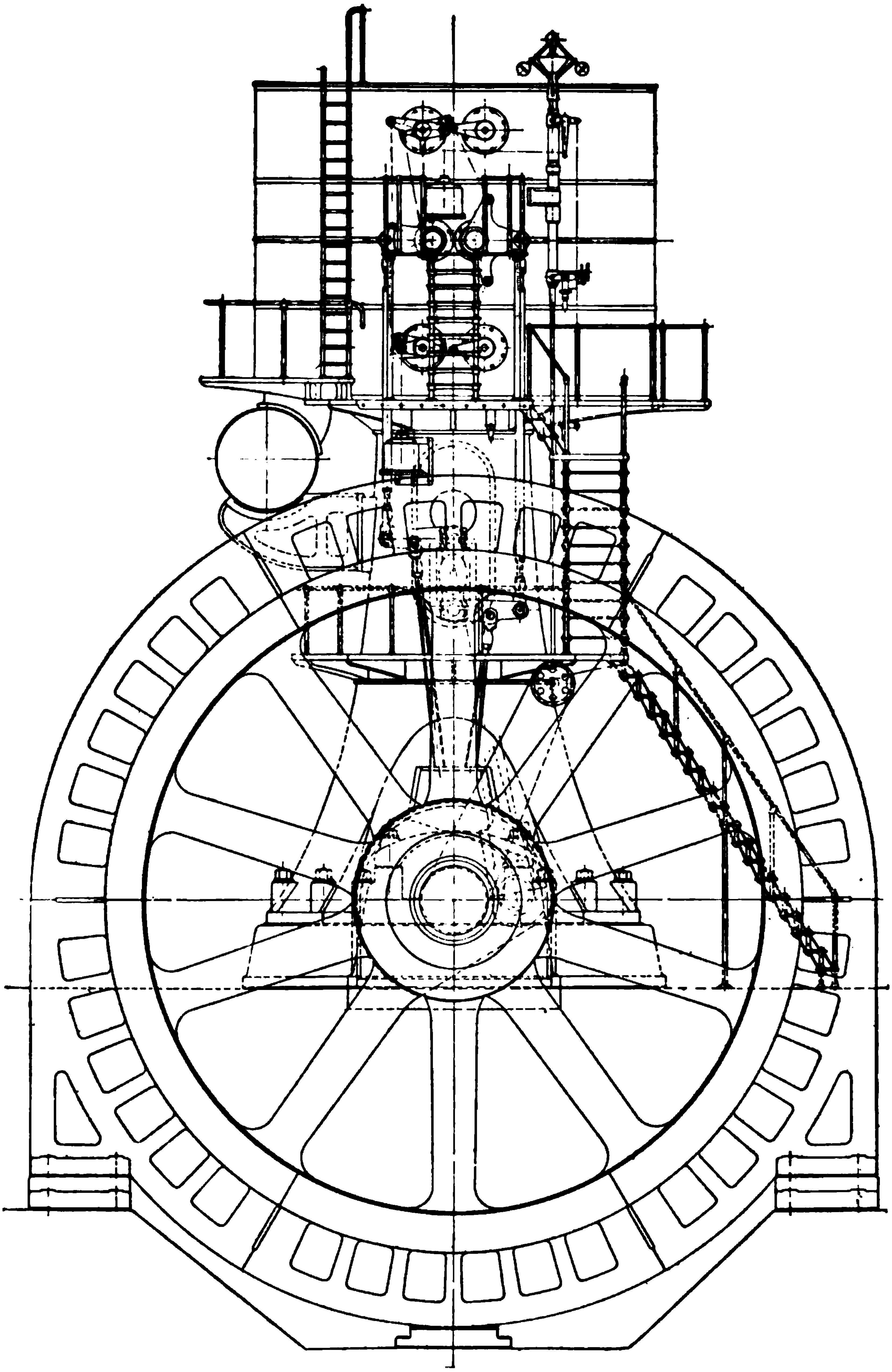


FIG. 150a. 3500 K.W. Vertical Cross-Compound Corliss Engine as Installed at the Power House of the Twin City Rapid Transit Co., Minneapolis, Minn.

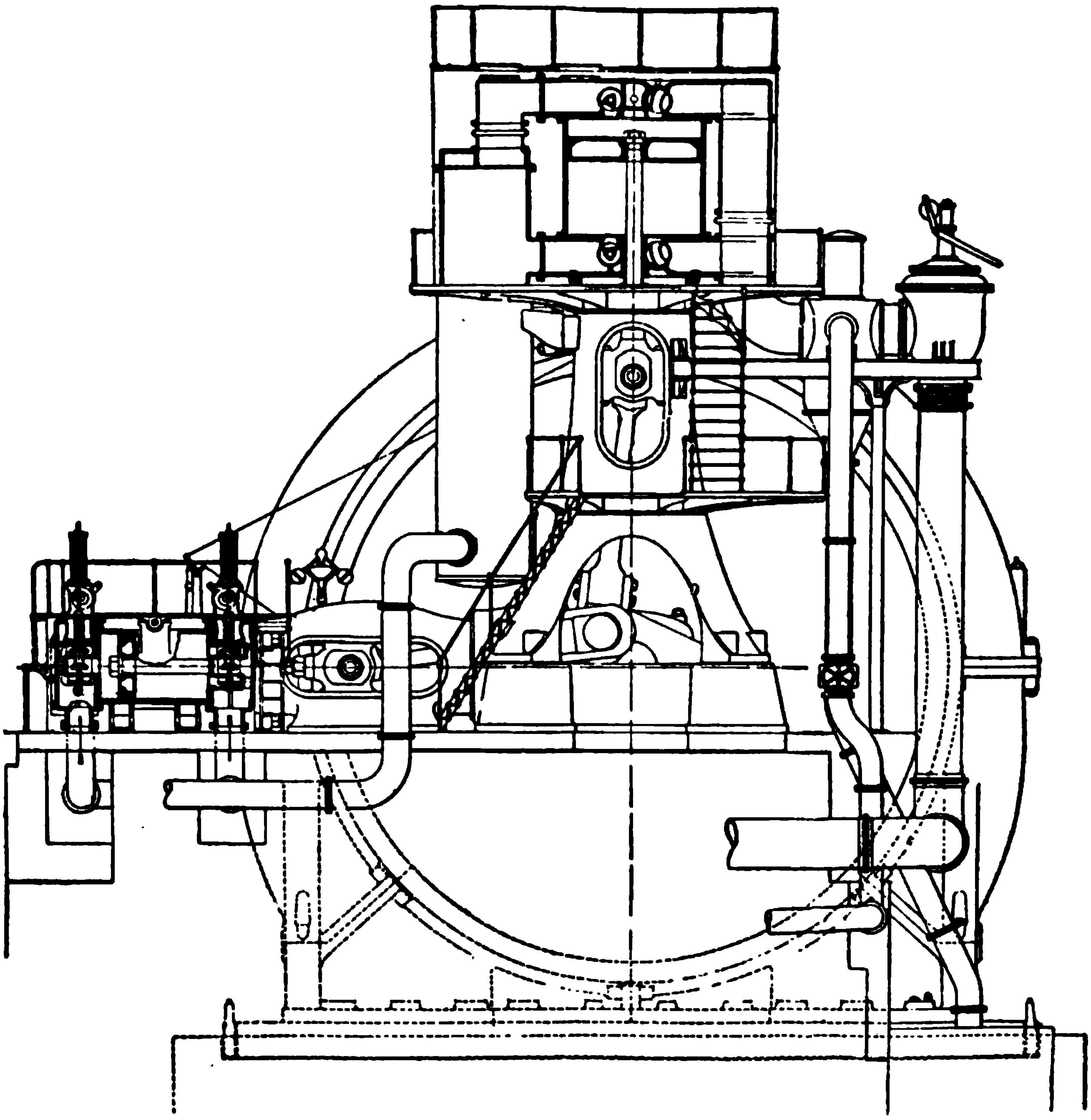


FIG. 150b. 7500 K.W. Vertical-Horizontal Double Compound Engine as Installed at the 59th Street Station of the Interborough. (Manhattan Type.)

The respective advantages and disadvantages of compounding may be tabulated as follows:

ADVANTAGES.	DISADVANTAGES.
1. Permits high range of expansion.	1. Increased first cost due to multiplication of parts.
2. Decreased cylinder condensation.	2. Increased bulk.
3. Decreased clearance and leakage losses.	3. Increased complexity.
4. Equalized crank effort.	4. Increased wear and tear.
5. Increased economy in steam consumption.	5. Increased radiation loss.

The ratio of expansion for a multi-expansion engine is usually taken to be the product of the ratio of the volume of large to small cylinder divided by the fraction of the stroke at cut-off in the high-pressure cylinder. For example, a compound engine with cylinders 24, 48 x 48 cutting off at $\frac{1}{3}$ in the high-pressure cylinder has a nominal ratio of expansion of $4 \div \frac{1}{3} = 12$. The number of expansions at rated load in compound condensing engines varies widely, ranging from 10 to 33, with an average not far from 16.

The steam consumption shown by tests of a number of compound engines using saturated steam, condensing and non-condensing, is given in Table 40. For tests with superheated steam see Table 43.

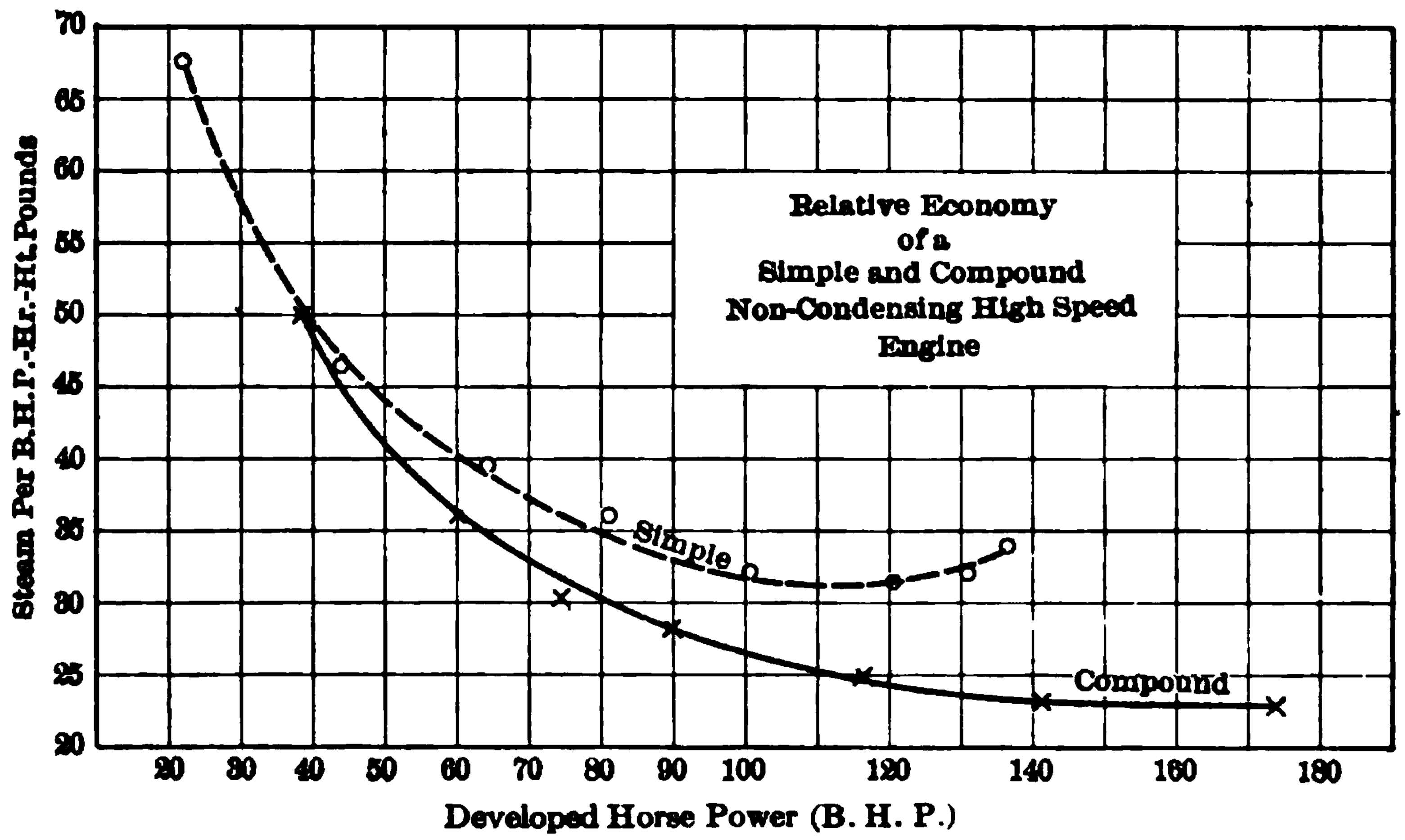


FIG. 151.

Fig. 151 shows the relative economy under comparable conditions of a high-speed simple and a high-speed compound engine, both running non-condensing and using saturated steam. The advantage of the compound at full load and overload is very marked, though its

economy drops off rapidly at light loads and may be less than that of the simple engine.

Fig. 152 shows the relative economy of two compound Corliss engines running condensing and non-condensing, both using saturated steam.

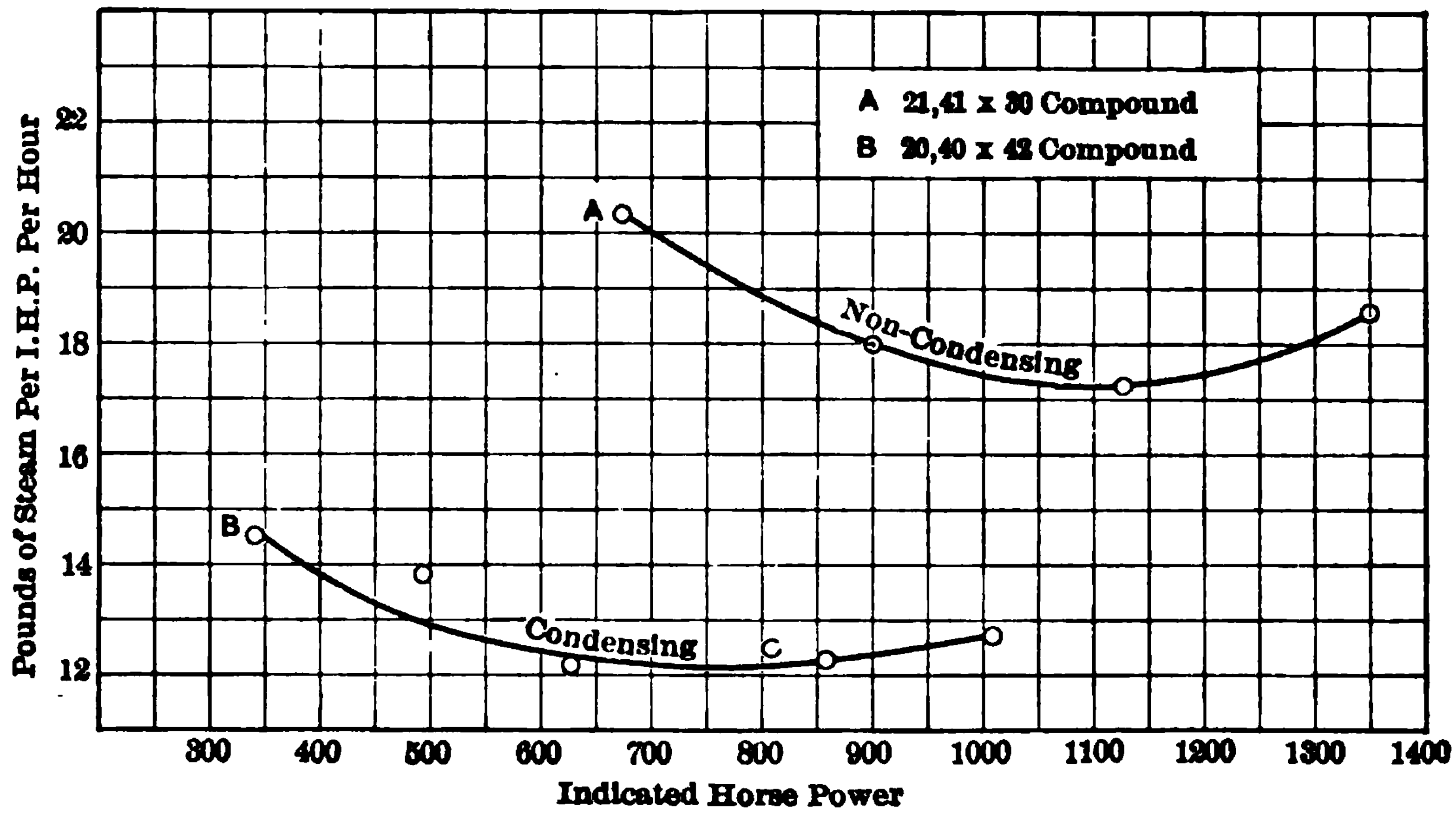


FIG. 152.

It should be borne in mind that the object of compounding is to permit the advantageous use of high pressures and large ratios of expansion. Under proper conditions compounding may increase the economy at rated load about 20 per cent for non-condensing engines and 30 per cent for condensing engines.

An exceptional performance of a single-valve high-speed non-condensing compound engine is that of engine No. 20, Table 40. With initial gauge pressure of 128 pounds the steam consumption is 22.3 pounds per I.H.P. hour, corresponding to a heat consumption of 376 B.T.U. per I.H.P. per minute.

One of the best performances of a multi-valve high-speed compound non-condensing engine is that of engine No. 14, Table 40. With initial pressure of 175 pounds gauge the steam consumption at full load is 17.17 pounds per I.H.P. hour, corresponding to a heat consumption of 291 B.T.U. per I.H.P. per minute.

The 8000-horse-power vertical cross compound Corliss engines of the Interborough Rapid Transit system (No. 6, Table 40), probably hold the record for economy for compound engines without jackets and reheaters, using saturated steam. With initial pressure of 175 pounds gauge and absolute back pressure of 2.2 pounds, the steam consumption is 11.96 pounds per I.H.P. hour, corresponding to a heat

consumption of 220 B.T.U. per I.H.P. per minute. In estimating average practice it would be safe to add 10 per cent or 20 per cent to the steam consumptions given in Table 40.

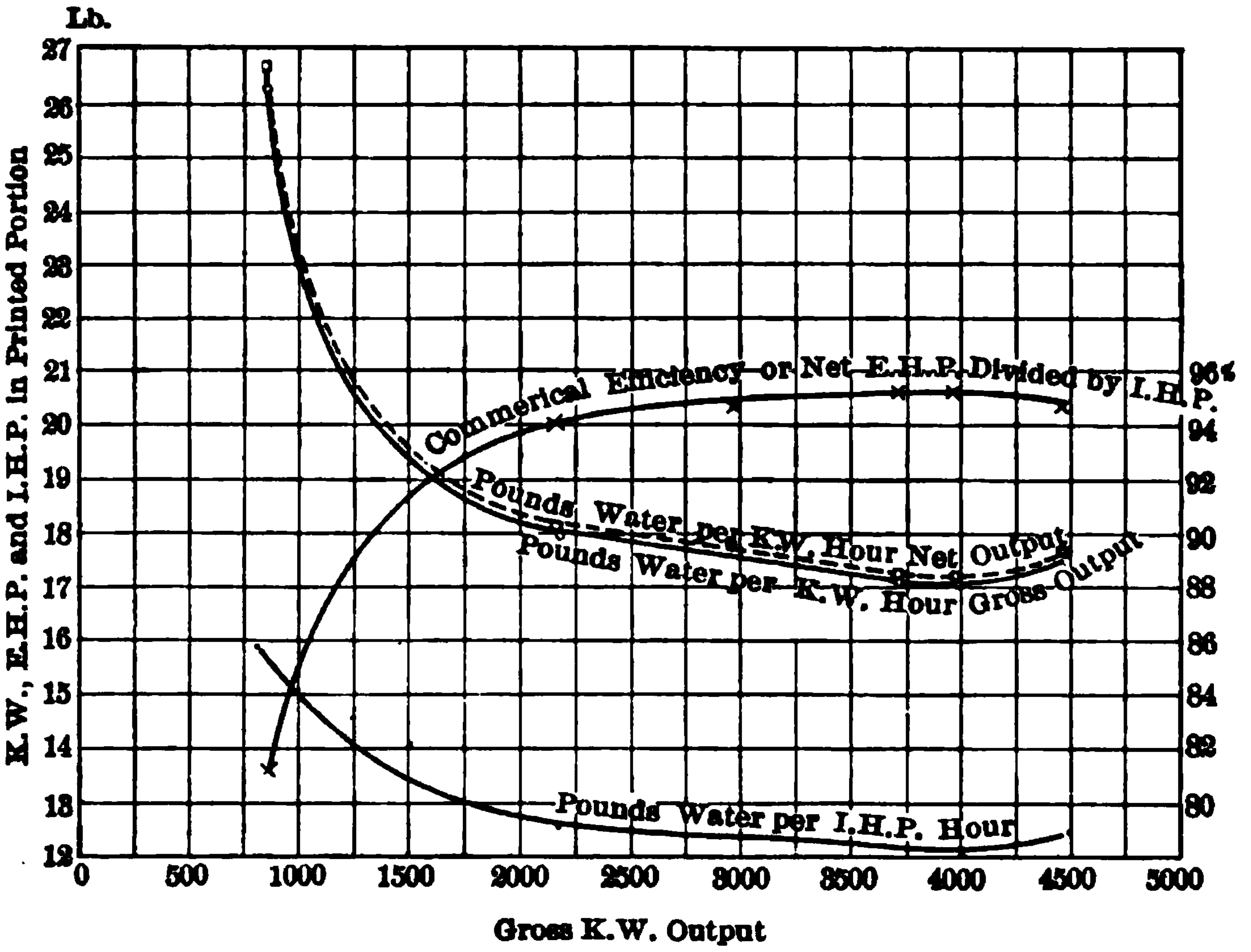


FIG. 153. Economy Test of the 5500-Horse-Power Three-Cylinder Compound Engine and Generator at the Waterside Station of the New York Edison Co.

Fig. 153 illustrates the performance of the 5500-horse-power three-cylinder compound engine at the Waterside Station of the New York Edison Company. The best economy is 11.93 pounds of steam per I.H.P. hour, corresponding to a heat consumption of 221 B.T.U. per I.H.P. per minute.

Compound Engines.— Best Load for Compound Engine: Trans. A.S.M.E., 18–674. *Cylinder Proportions for Compound and Triple Expansion Engines:* Trans. A.S.M.E., 21–1002, 16–762; Engr. U.S., Sept. 1, 1906, p. 586; Eng. News, March 2, 1899, p. 137; Eng. Rec., Jan. 7, 1899, p. 122; Power, June, 1904, p. 47. *Laws of the Average Simple vs. Compound Engines under Variable Load:* Am. Mach., Sept. 27, 1900, p. 927; *Non-condensing Compound Engine for Office Buildings:* Eng. Rec., June 18, 1898, p. 45. *Economical Use of Steam in Non-condensing Engines:* Eng. Mag., May, 1898, p. 213, July, 1898, p. 603.

Compound Engine Tests: Trans. A.S.M.E., 24–1274, 25–264; Engr. Lond., 99–546; Eng. News, Jan. 11, 1906, p. 44; Eng. Rec., April 16, 1898, p. 431, June 4, 1898, p. 1; Nov. 18, 1899, p. 579; Sibley Jour., May, 1901, p. 346; St. Ry. Jour., 27–41.

178. Triple and Quadruple Engines. — Triple and quadruple expansion engines are in general use where the load is practically constant, as in marine and pumping-station practice, but have been abandoned in street-railway work and in plants where the load fluctuates widely,

TABLE 40. — EXAMPLES OF STEAM-ENGINE ECONOMY.

MULTIPLE EXPANSION ENGINES, SATURATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions.	Cylinder Ratio.	Horse Power.	Initial Press., Lbs. Gauge.	Back Press., Lbs. Abs.	R.P.M.	M.E.P. Referred to L.P. Cyl.	Temp. Feed Water, Deg. F.	Lbs. of Steam per I.H.P. Hr.	B.T.U. per I.H.P. per Min.	Thermal Efficiency, per Cent.	B.T.U. per I.H.P. Min., Perfect Eng.	Efficiency Ratio, per Cent.	Mech. Efficiency, per Cent.	Lbs. of Coal per D.H.P. Hr.
	Quadruple Expansion.																
	Triple Expansion.																
	Compound, Condensing.																
6	Allis-Chalmers Engines, New York Subway.	Power, Feb., 1906, p. 115.	2(42),2(86) x 60	1:4.2	7365	175	2.2	75	27.9	130	11.96	220	19.2	159	72.4	†93.0	
7	Cross-Compound Corliss, Atlantic Mills, Providence.	Am. Elecn., June, 1903, p. 260.	16, 40 x 48	1:6¼	500	170	0.8	80	20.5		11.20	222	19.0	141	63.5		1.30
8	Leavitt Pumping Engine, Louis-	Trans. A.S.M.E., Vol. 16, p. 169.	27, 54 x 120	1:4	643	187	0.85	18.6	24.9	100	12.20	222	19.0	150	67.6	96.0	
9		Trans. A.S.M E., Vol. 24, p. 1274.	20, 40 x 42	1:4	627	151	0.85	121	19.4	121	12.10	222.7	19.0	143	64.3		
10		Trans. A.S.M E., Vol. 25, p. 212.	15, 40½ x 27	1:7.3	348	150	2.0	152	13.0	126	12.53	225.8	16.7	162	71.7		
11		Power, Oct., 1903, p. 583........	19, 34 x 30	1:3¼	340	100	2.0	150	16.5	126	12.60	229	18.5	175	76.5		
12		Barrus, Eng. Tests, p. 165......	18, 44 x 72	1:6.4	6803	145.2	1.5	60.3	20.6	116	12.70	234	18.1	157	67.0		
13	Edison Waterside Sta., N. Y. .	Power, July 1904, p. 424.. ...	43½, 75.3 x 60	1:6.02	5442	185	1.5	76.3	26.5	116	11.93	221	19.2	150	68.0	†95.2	
	Compound, Non-Condensing.																
			21, 41 x 30	1:3.8	1125	175	Atmos.	120	47.0	Assumed for Comparison, 212° F.	17.17	291	14.5	229	78.8		
			10, 14 x 6	1:2	39.6	165.0	Atmos.	401	42.4		19.2	328	12.8	234	71.5		
			10, 14 x 6	1:2	33.0	113.9	15.7	402			21.4	358	12.4	297	69.0		
			12, 20 x 13	1:2.8	187.5	169.6	Atmos.	271	34.8		21.14	357	11.7	233	65.5	93.5	
			17, 27 x 24	1:2½	540	148	Atmos.	210	37.1		19.3	328	13.0	244	75.0	†87.5	
			16, 27 x 18	1:2.84	375	130	Atmos.	226	31.0		21.14	355	11.9	255	72.0	91.5	
			12, 18 x 10	1:2¼	121	128	Atmos	271	35.0		22.3	376	11.2	258	68.5	93.0	
			12, 20 x 14	1:2.77	185	150	Atmos.		56.0		20.9	354	11.9	242	68.5	92.0	
			17½, 28 x 48	1:2.56	486.7	125.3	Atmos	99	32.9		21.59	362	11.7	260	71.9		
			14.2, 23.7 x 25.2	1:2.78	495	210	Atmos	80	55.0		18.6	316	13.4	216	68.5	91.0	

* Combined efficiency of pump and engine. † Combined efficiency of engine and generator.

in favor of the two or three-cylinder compound. The best economy on a *heat-unit* basis ever recorded for an engine using saturated steam was that of the Nordburg quadruple pumping engine at Wildwood, Pa., which gave a consumption of 12.26 pounds per I.H.P. hour and a heat consumption of 186 B.T.U. per I.H.P. per minute reckoned above the feed-water temperature.* The Allis triple expansion pumping engine at Chestnut Hill holds the record for saturated steam consumption, 10 pounds per I.H.P. hour, and its exceptional performance of one developed horse power per 1.09 pounds of coal has, perhaps, never been excelled. An inverted vertical marine cross compound engine, 21 and 36 x 36, built by Cole, Marchent & Morley, Bradford, England, holds the record for superheated steam consumption, 8.58 pounds per I.H.P. hour. (Table 47.) On the heat basis (192 B.T.U. per I.H.P. per minute), however, it does not equal the performance of the Nordburg engine. The above efficiencies have been exceeded by the binary vapor engine at Berlin; but this belongs in a class by itself and should hardly be compared with the ordinary form of steam engine. (See paragraph 182.)

Triple Expansion Engines. — Cylinder Proportions for Triple Expansion Engines : Trans. A.S.M.E., 21-1002, 10-576. *Economy of Triple Expansion Engines :* Trans. A.S.M.E., 8-496.

179. Effects of Condensing. — The effect of the condenser upon the power and economy of engines is indicated in Table 41. The curves in Figs. 154 and 155 were plotted from tests made by Professor R. L. Weighton on a 7, 10½, 15½ x 18 triple expansion engine at Durham College of Science, Newcastle-on-Tyne. The straight line shows how the mean effective pressure would vary with the degree of vacuum if the power increased directly with the reduction in back pressure. The curved line shows the actual M.E.P., which increases almost along the theoretical line up to a 10-inch vacuum, from which point on the increase is less marked. At 26 inches the actual M.E.P. reaches an apparent maximum. These figures are not applicable to all engines but give a good idea of the limitation of the vacuum with the reciprocating engine. The gain in steam consumption due to the condenser does not indicate a corresponding gain in heat consumption. For example, Engine No. 2, Table 41, shows an apparent gain in steam consumption, due to condensing, of 12.5 per cent, the temperature of the feed water returned to the boiler being 120 degrees F. With a suitable heater the exhaust of the non-condensing engine would be capable of heating

* Replaced in 1905 by a Riedler pumping engine on account of high maintenance cost.

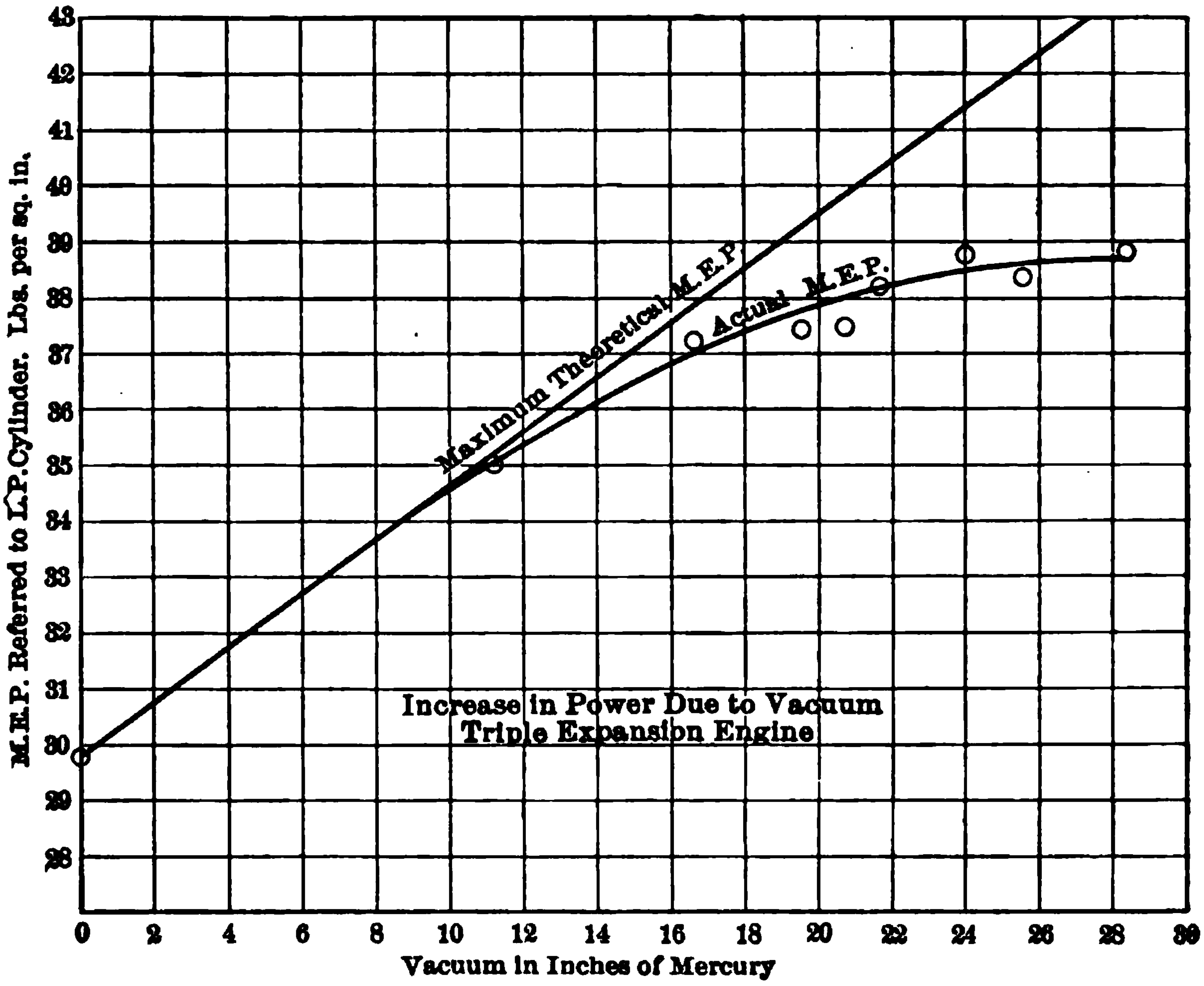


FIG. 154.

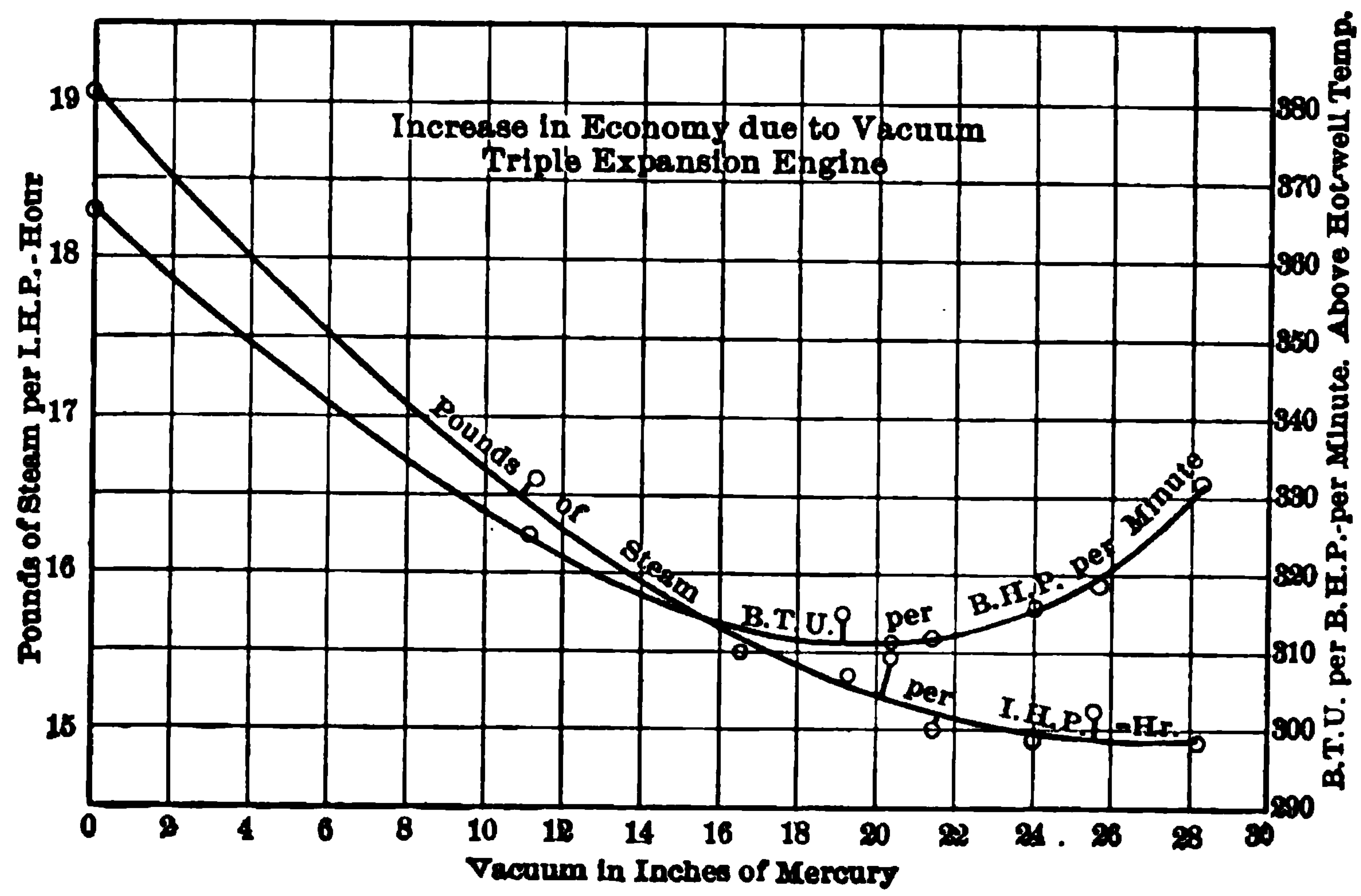


FIG. 155.

the feed water to 210 degrees F. The non-condensing engine should therefore be credited with 210 – 120 or 90 heat units per pound of steam used, or, in round numbers, 9 per cent. The difference between 12.5 per cent and 9 per cent, or 3.5 per cent, represents the net gain in favor of condensing provided the power necessary to create the vacuum is ignored. Actually the steam consumption of the condenser pumps might be equal to or greater than 3.5 per cent of the steam generated and the net gain becomes zero or even negative. Referring to Fig. 155, plotted from tests of the 7, 10½, 15½ x 18 triple expansion engine mentioned above, the solid lines show the feed-water consumption per I.H.P. hour and the broken line the heat units consumed per brake

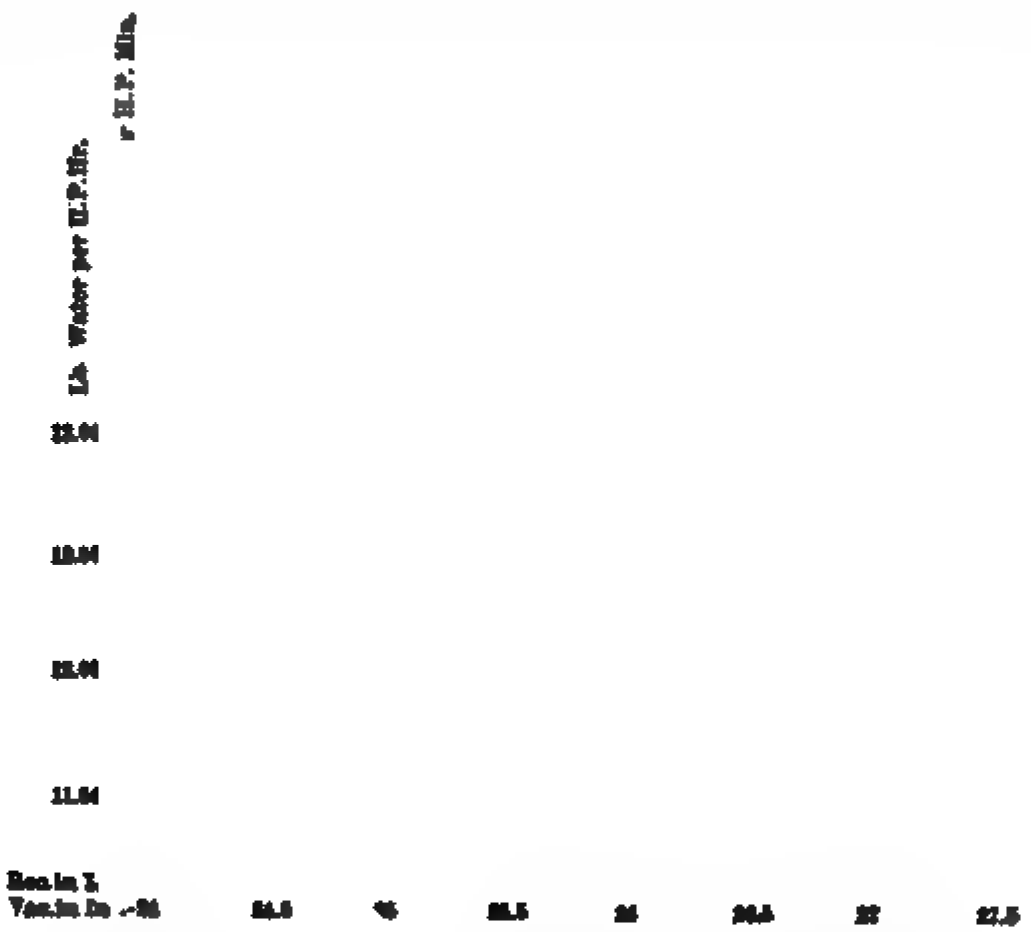


Fig. 156. Performance of 5500 H. P. Engine at Waterside Station of New York Edison Company.

horse power per minute measured above the hot-well temperature. The engine efficiency, based upon the water consumption, increases as the vacuum increases, reaching a maximum between 26 and 28 inches, whereas the heat-unit curve gives the maximum between 20 and 21 inches. Between 22 and 28 inches the heat-unit curve shows a rapid falling off in economy. Tests of the 5500-horse-power engine at the New York Edison Company's Waterside Station showed that increasing the vacuum from 25.3 to 27.3 inches decreased the water rate only 0.06 pound per I.H.P. (*Power*, July, 1904, p. 424) The results are illustrated in Fig. 156. In most cases, and particularly with large

compound engines, the net gain due to condensing is considerable, but the feed-water temperatures and power consumed by the auxiliaries should be taken into account.* Fig. 149 shows the effect of vacuum on the steam consumption of a small high-speed simple engine, and Fig. 152 of a cross compound Corliss. (See also paragraph 210.)

TABLE 41.

EXAMPLES OF THE EFFECT OF CONDENSING ON THE ECONOMY OF RECIPROCATING ENGINES.

	Non-Condensing.			Condensing.				Increase Due to Condensing.	
Reference Number.	Initial Gauge Pressure.	Horse Power Developed.	Steam Consumption, Pounds per H.-P. Hour.	Initial Gauge Pressure.	Back Pressure, Pounds per Square Inch Abs.	Horse Power Developed.	Steam Consumption, Pounds per H.-P. Hour.	In Power, per Cent.	In Economy, per Cent.
1	147	54.7	19.2	149	1.6	83.4	14.8	52.5	25
2	148	540	19.3	147	4	......	16.9	*	12.5
3	126	83	23.8	130	7.4	116	19.1	39.8	19.7
4	67.6	209	28.9	67	4.5	213	22	1.9	23.5
5	103.8	177.5	22.1	103.8	1.2	155	16.5	*	25.1
6	114	160	31	114	......	168	27	2	12.9
7	96	120	23.9	96	4	145	19.4	20.8	18.8
8	118	267	23.24	119	4.2	276.9	16	3.7	31
9	75.9	310	25.6	79	6.4	336	20.5	8.7	19.9
10	62.5	451	30.1	63.6	7.8	444	23	*	23.6
11	186.7	40.4	18.7	184.6	1.6	29.8	12.7	*	32

* Cut-off changed for best economy.

1. 7, 10½, 15½ x 18 triple; Eng. News, Aug. 21, 1902, p. 127.
2. 17, 27 x 24 Westinghouse marine, non-condensing; Power, August, 1903.
3. 1, 18 x 10 Buffalo tandem compound; Elec. World, Sept. 10, 1904, p. 404.
4. 18 x 30 four-valve (slide); Engine Tests, Barrus, p. 88.
5. 21, 65 x 43.31 Corliss; Peabody's Thermodynamics, p. 382.
6. 12 x 12 Reeves simple; Elec. World, Oct. 1, 1904, p. 587.
7. 18 x 48 simple Corliss; Peabody's Thermodynamics, p. 354.
8. 14, 28 x 24 two-valve (slide); Engine Tests, Barrus, p. 175.
9. 17 x 24 two-valve; Engine Tests, Barrus, p. 70.
10. 28 x 36 Corliss; Engine Tests, Barrus, p. 97.
11. Willans triple expansion central valve engine; Peabody, Thermodynamics, p. 406.

180. Throttling *vs.* Automatic Cut-Off. — The action of the governor in the throttling engine is shown by the superposed indicator cards (Fig. 157) taken between zero or friction load and maximum load. The effect of throttling is to reduce the pressure during admission, but does not change the point of cut-off or other events of the stroke. The steam may be partially dried or even superheated by throttling, thus tending to reduce cylinder condensation. Initially dry saturated steam at a pressure of 125 pounds gauge would be super-

* See *Power*, Feb. 23, 1909, p. 381.

heated about 12 degrees in expanding through a throttle to 90 pounds, or if it contained initially 2 per cent moisture would be perfectly dried in expanding to 40 pounds. (See Table 42.) Friction through the valve also tends to dry the steam. Thus with very light loads the superheat may be decidedly appreciable. The possible gain due

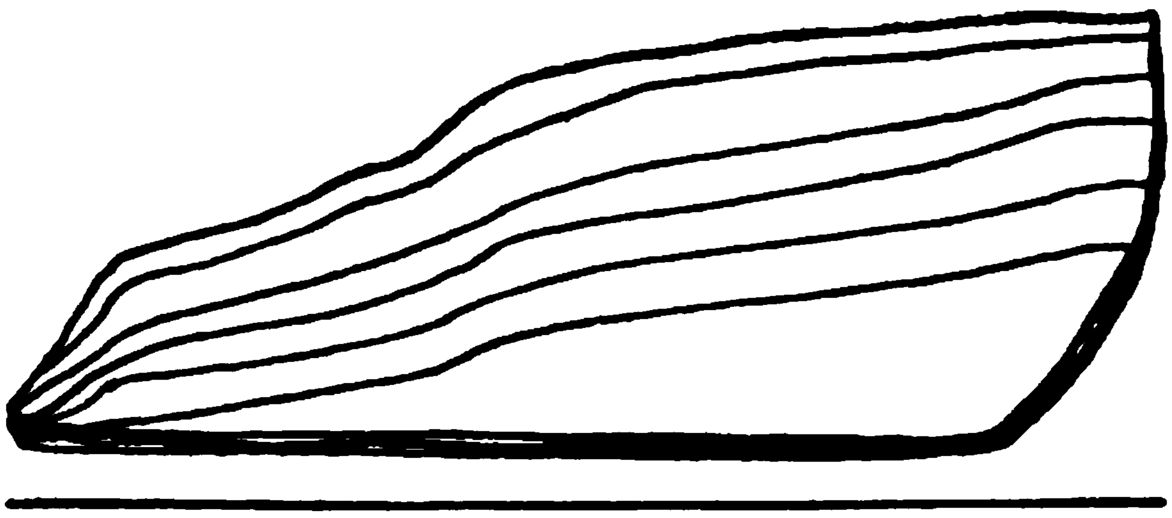


FIG. 157. Typical Indicator Cards. High-Speed Throttling Engine.

to decreased cylinder condensation is to some extent offset by incomplete expansion. The best efficiency for a given load is realized by a proper compromise between cut-off and initial pressure. Experiments made by Professor Denton (Trans. A.S.M.E., 2–150) on a 17 x 30 non-condensing double-valve engine showed the most economical results with $\frac{1}{4}$ cut-off for 90 pounds pressure, $\frac{1}{3}$ cut-off for 60 pounds, and $\frac{45}{100}$ for 30 pounds. The average throttling engine does not give close regulation, the governor usually lacking sensitiveness. Tests show the economy to be better than that of the automatic engine on light loads, and the crank effort more uniform.

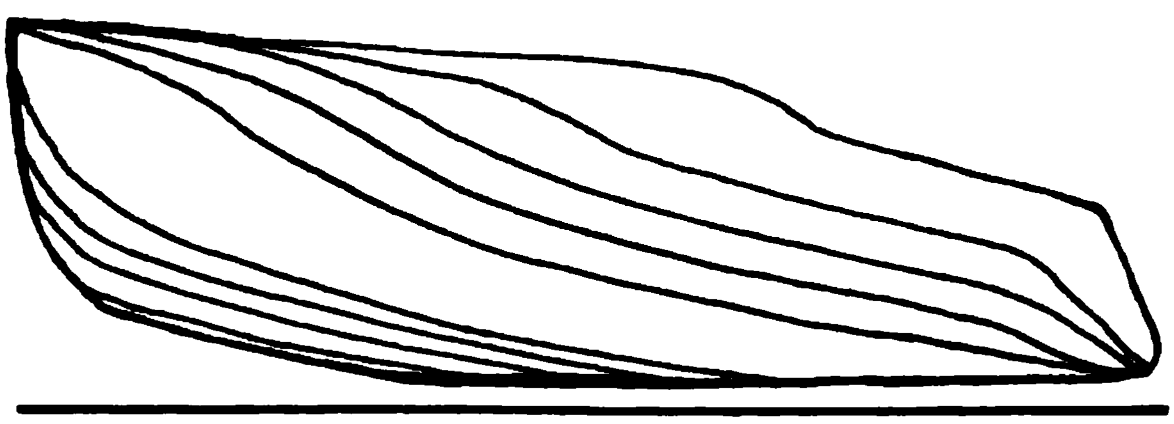


FIG. 158. Typical Indicator Cards. High-Speed Automatic Engine.

The indicator cards shown in Fig. 158 were taken from a single-valve high-speed automatic engine operating between friction load and maximum load. The mean effective pressure is adjusted to suit the load by the automatic variation in the cut-off, the initial pressure remaining the same. Since the cut-off is controlled by the action of the governor on the single valve, all other events of the stroke are

likewise changed. With a four-valve engine the variation in cut-off does not affect the other events.

The chief advantage of the automatic over the throttling engine lies in its sensitive regulation, and while, in general, it gives a lower steam consumption than the throttling engine, this is probably in most cases due to superior construction and not to the method of governing.

TABLE 42.

SHOWING THE INITIAL PER CENT OF MOISTURE THAT WILL BE EVAPORATED IN THROTTLING FROM A HIGHER TO A LOWER PRESSURE.

Based on Marks' and Davis' Steam Tables.

Final Pressures.	Initial Pressure, Absolute.								
	80	85	90	95	100	105	110	115	120
80	......	0.13	0.24	0.36	0.45	0.55	0.65	0.74	0.83
75	0.14	0.26	0.37	0.49	0.59	0.70	0.79	0.88	0.97
70	0.28	0.40	0.52	0.64	0.74	0.84	0.93	1.03	1.12
65	0.43	0.55	0.66	0.78	0.88	0.99	1.08	1.18	1.26
60	0.59	0.71	0.83	0.95	1.06	1.16	1.25	1.34	1.44
55	0.77	0.89	1.01	1.13	1.23	1.34	1.44	1.53	1.62
50	0.97	1.09	1.21	1.33	1.43	1.54	1.64	1.74	1.82
45	1.19	1.32	1.44	1.56	1.66	1.76	1.86	1.96	2.05
40	1.44	1.56	1.68	1.80	1.91	2.02	2.12	2.21	2.30
35	1.72	1.85	1.97	2.10	2.20	2.31	2.41	2.51	2.60
30	2.05	2.18	2.30	2.42	2.53	2.64	2.74	2.84	2.93
25	2.44	2.56	2.69	2.82	2.92	3.03	3.13	3.23	3.32
20	2.90	3.04	3.16	3.29	3.40	3.51	3.61	3.71	3.80
15	3.51	3.65	3.78	3.90	4.01	4.13	4.23	4.33	4.43

Final Pressures.	Initial Pressure, Absolute.								
	125	130	135	140	145	150	155	160	165
80	0.91	0.99	1.08	1.15	1.22	1.29	1.35	1.41	1.48
75	1.05	1.13	1.21	1.28	1.36	1.43	1.49	1.55	1.62
70	1.19	1.27	1.36	1.43	1.50	1.58	1.64	1.70	1.77
65	1.34	1.43	1.51	1.59	1.66	1.73	1.79	1.85	1.93
60	1.52	1.60	1.68	1.76	1.83	1.90	1.96	2.03	2.10
55	1.70	1.78	1.86	1.94	2.02	2.09	2.15	2.21	2.29
50	1.90	1.99	2.08	2.15	2.22	2.30	2.36	2.42	2.50
45	2.13	2.21	2.30	2.38	2.45	2.52	2.59	2.65	2.73
40	2.39	2.47	2.55	2.63	2.71	2.78	2.84	2.91	2.99
35	2.68	2.77	2.85	2.93	3.01	3.08	3.14	3.21	3.29
30	3.01	3.10	3.18	3.26	3.34	3.41	3.48	3.55	3.63
25	3.41	3.49	3.58	3.66	3.74	3.81	3.88	3.96	4.04
20	3.88	3.97	4.06	4.15	4.22	4.30	4.37	4.45	4.53
15	4.51	4.60	4.70	4.78	4.86	4.94	5.01	5.09	5.17

The following performances of a Belliss 250-horse-power high-speed condensing engine fitted with both automatic and throttling governing devices give results decidedly in favor of the throttling engine. (Pro. Inst. of Mech. Engrs., 1897, p. 331.)

	Automatic Cut-Off.				Throttling.			
Percentage of load	100	62.5	33	25	100	62.5	33	25
Electrical horse power	213	132	77.8	53	213	132	77.8	53
Steam per I.H.P. hour	22.5	22.9	28.5	34.3	21	21.7	25.6	28.4

Some of the comparative advantages and disadvantages of the automatic and throttling engines are as follows:

Advantages.

AUTOMATIC.

1. Sensitiveness of regulation.
2. Increased ratio of expansion.
3. Low terminal pressures.

THROTTLING.

1. Low first cost.
2. Crank effort more uniform.
3. Reduced cylinder condensation.
4. Simplicity of regulating device.

Disadvantages.

AUTOMATIC.

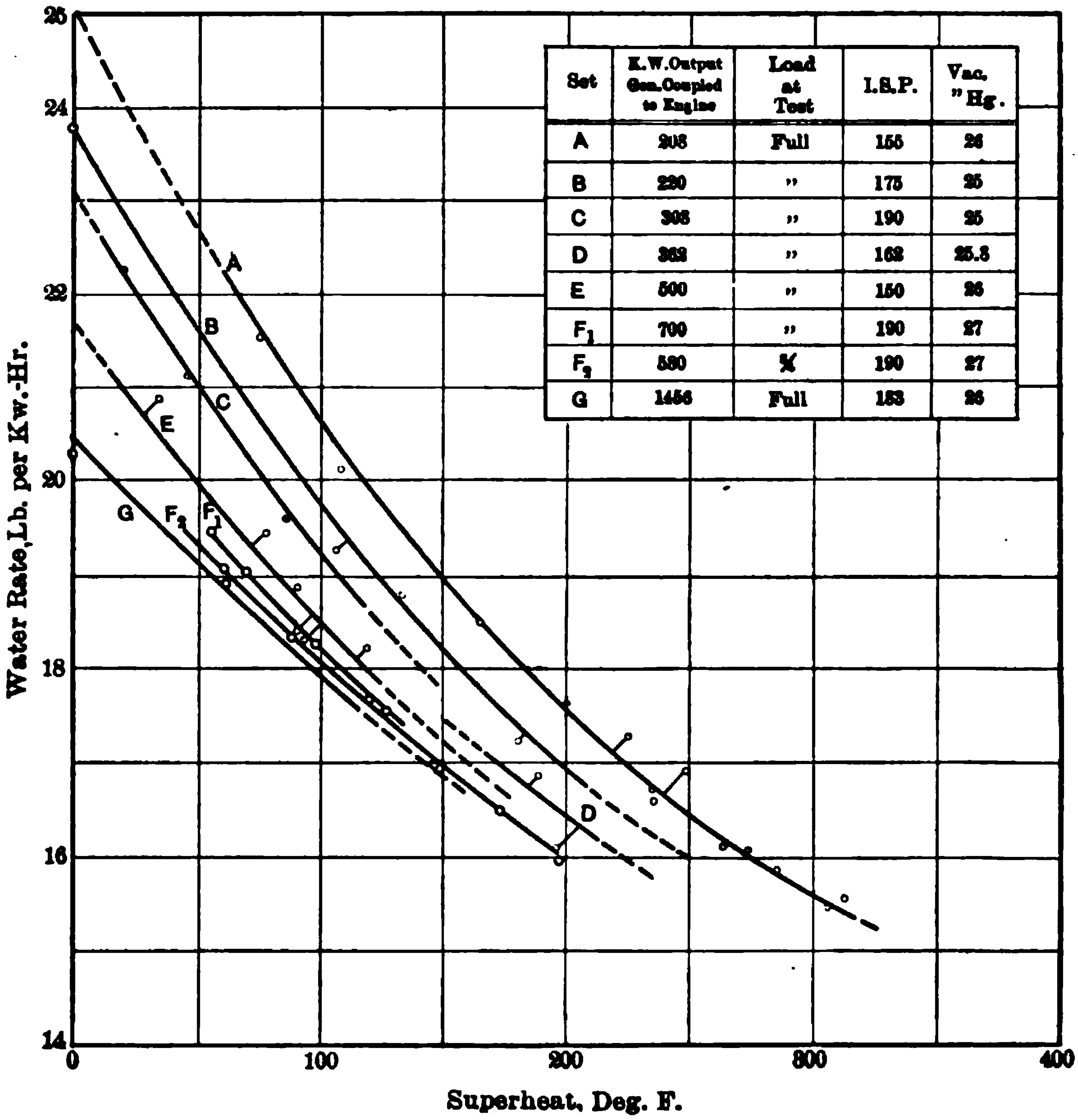
1. Increased cylinder condensation.
2. Greater variation in crank effort.
3. Complicated valve gear.
4. Low economy at very early loads.

THROTTLING.

1. Low ratio of expansion.
2. High terminal pressure.
3. Low initial pressure at early loads.

181. Influence of Superheat. — (See also paragraph 103.) Table 43 gives test results for several different types of engines employing superheated steam. These figures may be compared with the performances of engines using saturated steam as given in Tables 39 and 40. A decided gain in economy is shown in favor of superheat for single-cylinder engines. With compound engines the advantage is not so apparent, while triple expansion engines show the least gain. Tables 44 to 46 show the effect of superheating on simple, compound, and triple expansion engines. (Proc. A.S.M.E., September, 1907.) As far as steam consumption is concerned, most engines show greater economy with superheated than with saturated steam, but the gain in thermal efficiency is not so marked, and when the economy is measured in dollars and cents per developed horse power, taking all things into consideration the gain is still further reduced and in many cases completely neutralized.

Fig. 159 gives the results of a series of tests made on a number of Belliss & Morcom engines using superheated steam. (Pro. Inst. of Mech. Engrs., March, 1905, p. 302.) The engines were from 200 to 1500 kilowatts capacity and were tested at full load. It is noticeable that the curves all converge to a single point and will meet at about 400



Set	K.W. Output Gen. Coupled to Engine	Load at Test	I.S.P.	Vac. " Hg.
A	208	Full	155	26
B	220	"	175	25
C	308	"	190	25
D	362	"	162	25.8
E	500	"	150	26
F_1	700	"	190	27
F_2	580	¾	190	27
G	1456	Full	153	26

FIG. 159. Effect of Superheat on Steam Consumption.

degrees F. The results show that if sufficient superheat is put into the steam all engines of whatever size are equally economical. Fig. 160 shows the relationship between degree of superheat and the heat consumption at various loads for a 300-horse-power Belliss & Morcom high-speed triple expansion engine. (Pro. Inst. Mech. Engrs., March, 1905, p. 303.) It will be noted that the variation in heat consumption at different percentages of load becomes less marked as the degree of superheat increases. With superheat of 350 degrees F. the heat consumption from ½ load to full load is practically constant.

TABLE 48. — EXAMPLES OF STEAM-ENGINE ECONOMY.

SUPERHEATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions, Inches.	Cylinder Ratio.	Horse Power.	Initial Press., Lbs. Gauge.	Back Press., Inches Mercury.	R.P.M.	Temp. Hot Well, Deg. F. *	Lbs. of Steam per I.H.P. Hr.	B.T.U. per I.H.P. per Min. †	Thermal Eff., per Cent.	B.T.U. per I.H.P., Min., Perfect Eng.	Efficiency Ratio, per Cent.	Superheat at Admission, Deg. F.	Temp. Steam at Admission, Deg. F.	Spec. Heat of the Superheated Steam. ‡
	Triple Expansion.																
	Compound.																
	Simple.																
16	Poppet-Valve, Condens'g.	Zeit. d. V.D.I., Aug., 1905, p. 1310.	16.3 x 39.4		123	145	1.4	81.1	91	16.70	326.2	13.0	146	46.0	73.8	424	0.57
17	Poppet-Valve, Condens'g.	Zeit. d V.D.I., Aug., 1905, p. 1310.	16.3 x 39.4		120	145	1.5	81.2	92	14.70	307.0	13.6	146	47.6	226.2	576	0.55
18	Poppet-Valve, Non-Condensing.	Zeit. d. V.D.I., Aug., 1905, p. 1310.	16.3 x 39.4		123	145	Atmos.	81.5	212	16.10	307.5	13.8	242.2	79.0	254.3	604.3	0.64

* Ideal feed-water temp. † Later figures (June, 1906) give somewhat lower values. See Proc. A.S.M.E., May 14, 1906. ‡ Above ideal feed-water temp.

TABLE 44.

STEAM AND COAL SAVING IN A SIMPLE NON-CONDENSING ENGINE OF 250 I.H.P. WITH SUPERHEATED STEAM AT DIFFERENT TEMPERATURES.

Press. 12 atm. = 177 lb.; temp. of sat. steam 369 deg. F.; cut-off 20 per cent; piston speed 10 ft. per sec.; slide or piston valve; change of cut-off effected by valve gear.

Kind of Superheat.	Degrees of Superheat.	Temperature of Steam.	Cut-off Constant. I.H.P. Variable.							Cut-Off Variable. I.H.P. Constant.								Remarks.
			I.H.P.	At Engine.			At Boiler.			I.H.P.	At Engine.			At Boiler.			Cut-off, per Cent.	
				Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat Steam.		Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam.			Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam.		Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam.			
					Steam, per Cent.	Fuel, per Cent.		Feed Water, per Cent.	Fuel, per Cent.			Steam, per Cent.	Fuel, per Cent.		Feed Water, per Cent.	Fuel, per Cent.		
	Deg. F.	Deg. F.																
None	0	369	250	27.00	......	......	29.00	......	....	250	27.00	......	......	29.00	......	......	20.00	Indirect; superheater in boiler
75 to 150° F. Low:																		
Indirect	103	472	235	21.00	22.0	18.5	21.40	27	23	250	21.05	22.0	18.5	21.47	26.0	22.5	20.88	Direct; superheater separately fired
Direct	103	472	235	21.00	22.0	10.0	21.40	27	14	250	21.05	22.0	9.5	21.47	26.0	14.0	20.85	
150 to 225° F. Medium:																		
Indirect	162	531	230	19.85	26.5	20.5	20.10	30	27	250	20.00	25.5	20.0	20.40	29.5	24.0	21.35	
Direct	162	531	230	19.85	26.5	21.5	20.10	30	19	250	20.00	25.5	11.0	20.40	29.5	15.5	21.35	
225 to 290° F. High:																		
Indirect	234	603	222	18.50	31.0	23.0	18.85	35	31	250	18.75	30.5	22.0	19.13	34.0	26.0	21.75	
Direct	234	603	222	18.50	31.0	15.0	18.85	35	22	250	18.75	30.5	13.5	19.13	34.0	18.0	21.75	

TABLE 45.

STEAM AND COAL SAVING IN A COMPOUND ENGINE, CONDENSING, OF 250 I.H.P. WITH SUPERHEATED STEAM AT DIFFERENT TEMPERATURES.

Press. 10 atm. = 142.23 lbs.; temp. of sat. steam 354 deg. F., cut-off 6 per cent, piston speed 10 ft. per sec.; automatic cut-off; 4 poppet or piston valves per cyl.

Kind of Superheat.	Degrees of Superheat.	Temperature of Steam.	Cut-off Constant, I.H.P. Variable							Cut-off Variable, I.H.P. Constant								Remarks.
	Deg. F.	Deg. F.																
None	0	354																Indirect; superheater in boiler
75 to 150 deg. F.																		
Low: Indirect	130	484	225	12.25	15.0	11.0	12.50								12.70	20.0	16.0	Direct; superheater separately fired
Direct	130	484	225	12.25	15.0	1.5	12.50								12.70	20.0	7.0	
150 to 225 deg. F.																		
Medium: Indirect	202	556	215	11.60	21.0	15.0	11.81	25	21	250	7.50	11.75	20.0	14.0	11.98	24.5	18.5	
Direct	202	556	215	11.60	21.0	5.5	11.81	25	12	250	7.50	11.75	20.0	4.5	11.98	24.5	9.5	
Double: (1) Indirect, (2) Direct	202	556	205	10.70	27.5	18.0	10.93	31	24	250	8.00	10.85	26.0	17.0	11.09	30.0	21.0	
(1) Direct, (2) Indirect	202	556	205	10.70	27.5	16.5	10.93	31	23	250	8.00	10.85	26.0	15.0	11.09	30.0	19.5	
225 to 290 deg. F.																		
High: Indirect	274	628	205	10.70	27.5	19.0	10.93	31	27	250	8.00	10.85	26.0	17.5	11.09	30.0	21.5	
Direct	274	628	205	10.70	27.5	10.0	10.93	31	17	250	8.00	10.85	25.0	8.5	11.09	30.0	13.0	
Double: (1) Indirect, (2) Direct	274	628	198	10.27	30.0	18.5	10.50	34	26	250	8.50	10.51	28.5	16.0	10.75	32.0	20.5	
(1) Direct, (2) Indirect	274	628	198	10.27	30.0	17.0	10.50	34	25	250	8.50	10.51	28.5	15.0	10.75	32.0	19.0	
290 to 360 deg. F.																		
Very high: Indirect	338	692	198	10.27	30.0	20.5	10.50	34	28	250	8.50	10.51	28.5	18.5	10.75	32.0	22.5	
Direct	338	692	198	10.27	30.0	11.5	10.50	34	20	250	8.50	10.51	28.5	9.0	10.75	32.0	13.5	

TABLE 46.

STEAM AND COAL SAVING IN A TRIPLE EXPANSION ENGINE, CONDENSING, OF 5000 I.H.P. WITH SUPERHEATED STEAM AT DIFFERENT TEMPERATURES.

Press. 14 atm. = 199 lb.; temp. of sat. steam 381 deg. F.; cut-off 4 per cent; piston speed 16.5 ft. per sec.; automatic cut-off; 4 poppet or piston valves per cyl.

Kind of Superheat.	Degrees of Superheat.	Temperature of Steam.	Cut-off Constant. I.H.P. Variable.							Cut-off Variable. I.H.P. Constant.								Remarks.
				At Engine.			At Boiler.					At Engine.			At Boiler.			
			I.H.P.	Steam Consump. per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Steam, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	Feed Water, per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Feed Water, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	I.H.P.	Variation of Cut-off, per Cent.	Steam Consump. per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Steam, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	Feed Water per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Feed Water, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	
	Deg. F.	Deg. F.																
None	0	381	5000	11.82	......	......	12.70	......	...	5000	4.00	11.82	..		12 70	.....	.....	Indirect; superheater in boiler
75 to 150 deg. F.																		
Low: Indirect	144	525	4350	10 50	11.5	7.5	10.70	16	12	5000	4.85	10.90	7	3.0	11.25	11.5	7.5	Direct; superheater separately fired
Direct.	144	525	4350	10.50	11.5	−3.0	10.70	16	2	5000	4.85	10.90	7	−8.0	11.25	11.5	−2.5	
150 to 225 deg. F.																		
Medium: Indirect	216	597	4150	9.81	17.0	10.0	10.03	21	17	5000	5.15	10.32	13	6.0	10.52	17.5	11.0	
Direct.	216	597	4150	9.81	17.0	0.0	10.03	21	8	5000	5.15	10 32	13	−4 5	10.52	17.5	1.0	
Double: (1) Indirect (2) Direct	216	597	4000	9.15	22.5	12.5	9.35	26	19	5000	5.40	9.60	19	8 5	9.80	23.0	13.5	
(1) Direct (2) Indirect	216	597	4000	9.15	22.5	11.0	9 35	26	17	5000	5 40	9.60	19	6.5	9 80	23.0	12.0	
225 to 290 deg F.																		
High: Indirect	288	669	4000	9.15	22.5	13.0	9.35	26	21	5000	5.40	9.60	19	9.0	9.80	23.0	14.0	
Direct.	288	669	4000	9.15	22.5	4 0	9 35	26	12	5000	5 40	9.60	19	−1 0	9.80	23.0	5.0	
Double: (1) Indirect (2) Direct	288	669	3775	8.48	28.0	16.0	8 70	31	24	5000	5.85	9.20	22	9.0	9.40	26.0	14.5	
(1) Direct (2) Indirect	288	669	3775	8.48	28.0	14 5	8.70	31	22	5000	5 85	9.20	22	7.5	9.40	26 0	13.0	
290 to 360 deg. F.																		
Very high: Indirect	353	734	3775	8.48	28.0	18.0	8.70	31	25	5			22	11.5	9.40	26 [illegible]	16.5	
Direct.	353	734	3775	8.48	28.0	8.0	8.70	31	16	5			22	1 5	9.40	26 [illegible]	7.0	

FIG. 159a 3000 H.P. Sulzer Engine Designed for Highly Superheated Steam.

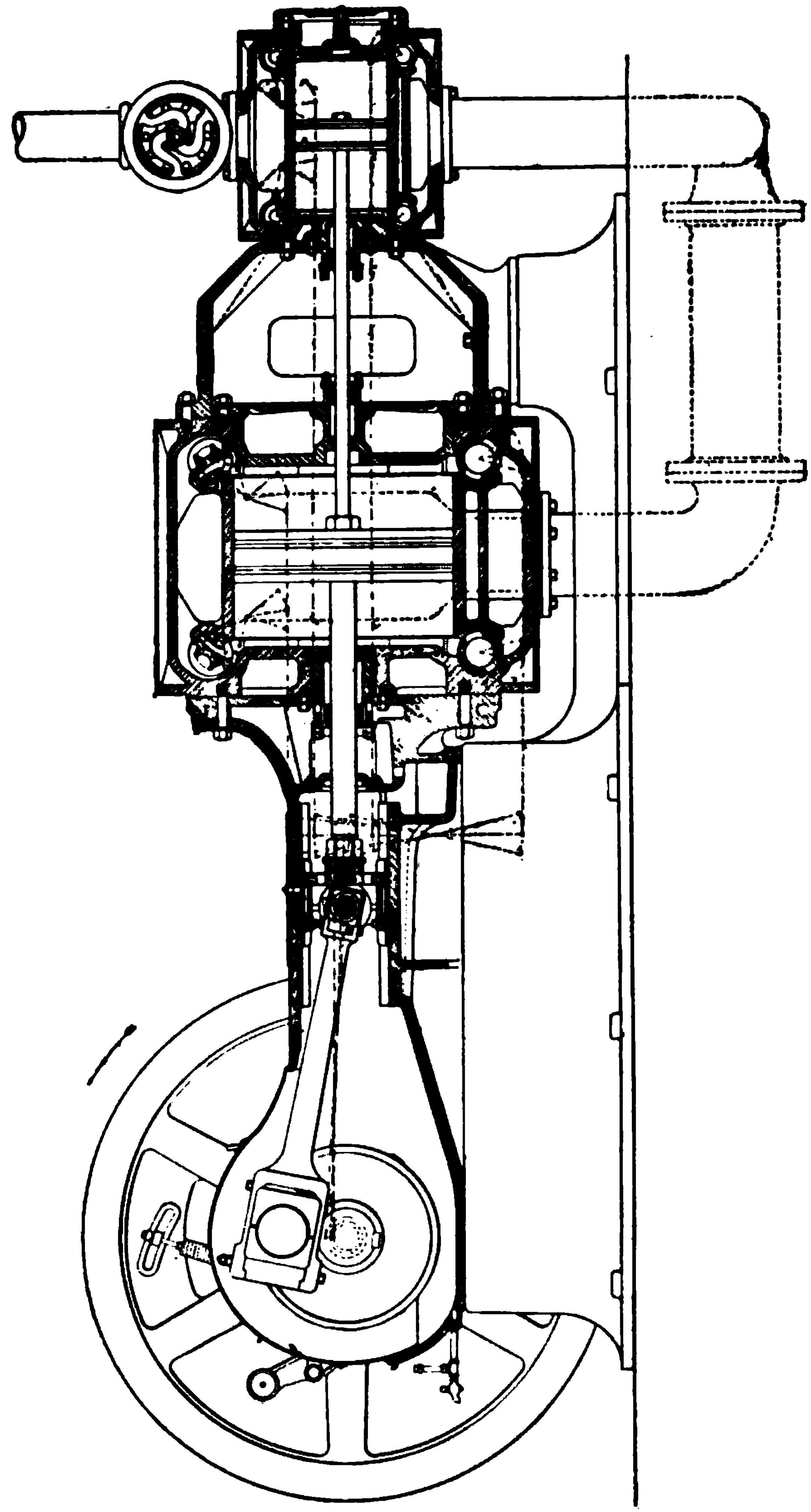


FIG. 159b. Fleming-Harrisburg Four-valve High Speed Tandem Compound Engine.

Table 47 gives the results of a test made on a 21, 36 x 36 inverted vertical marine engine. (*Engr., Lond.*, June 2, 1905, p. 546.)

TABLE 47.

PERFORMANCE OF 21, 36 × 36 INVERTED VERTICAL MARINE CROSS COMPOUND ENGINE.

(*Engr., Lond.*, June 2, 1905, p. 546.)

Pressure on boiler side of throttle valve, gauge	117.5	117.5	117.5	117	114.5	114.5
Temperature of steam on boiler side of throttle, degrees F.	743	738	749	751	732	726
Degrees superheat of steam on boiler side of throttle, F.	395	390	401	403	384	378
Temperature of steam at admission, F.	601	590	569	580	558	550
Degrees superheat of steam at admission, F.	253	242	221	232	210	202
Vacuum, inches of mercury, absolute	3.4	3	2.22	1.82	1.82	1.7
I.H.P.	481.3	461.1	347.5	333.5	258	145.52
M.E.P. referred to L.P. cylinder	26	29.9	18.8	18	14.4	7.87
Revolutions per minute	100.6	100.7	100.6	100.7	100.7	100.7
Pounds of steam per I.H.P. hour	9.09	9.26	8.88	8.68	8.74	8.58
B.T.U. per I.H.P. per minute	192.2	201.7	197.6	194	194	192.1
See footnote *	187	189	181	179	179	175
B.T.U. per I.H.P. per minute, perfect engine	142.4	142.5	130.2	126	128.5	128
Thermodynamic efficiency	21.4	21	21.4	21.8	21.8	22
Efficiency ratio	72	72	66	65	66	67
Equivalent evaporation of saturated steam reckoned from hot well	10.63	10.81	10.38	10.07	10.12	10.03
Temperature of hot well	102	101	78	71	64	70

* B.T.U. per I.H.P. per minute based on latest (May, 1908) values for specific heat of superheated steam.

The relationship between the weight of steam consumed per I.H.P. hour and the equivalent heat consumption of a 250-horse-power tandem compound Van Den Kerchove engine is illustrated in Fig. 161.

The performances of engines using superheated steam should be expressed in B.T.U. per I.H.P. per minute or the equivalent, as the steam consumption alone gives no idea of the true heat consumption.

182. Binary-Vapor Engines. — A consideration of the Carnot or Rankine cycles shows that theoretically the efficiency of the steam engine may be increased by raising the temperature of the steam supplied or by lowering the temperature of the exhaust, that is to say, by increasing the range. Superheated steam development has practically determined the upper limit, and economical practice indicates a vacuum of about 26 inches, corresponding to 126 degrees F., as the average lower limit for most efficient results from a commercial standpoint.

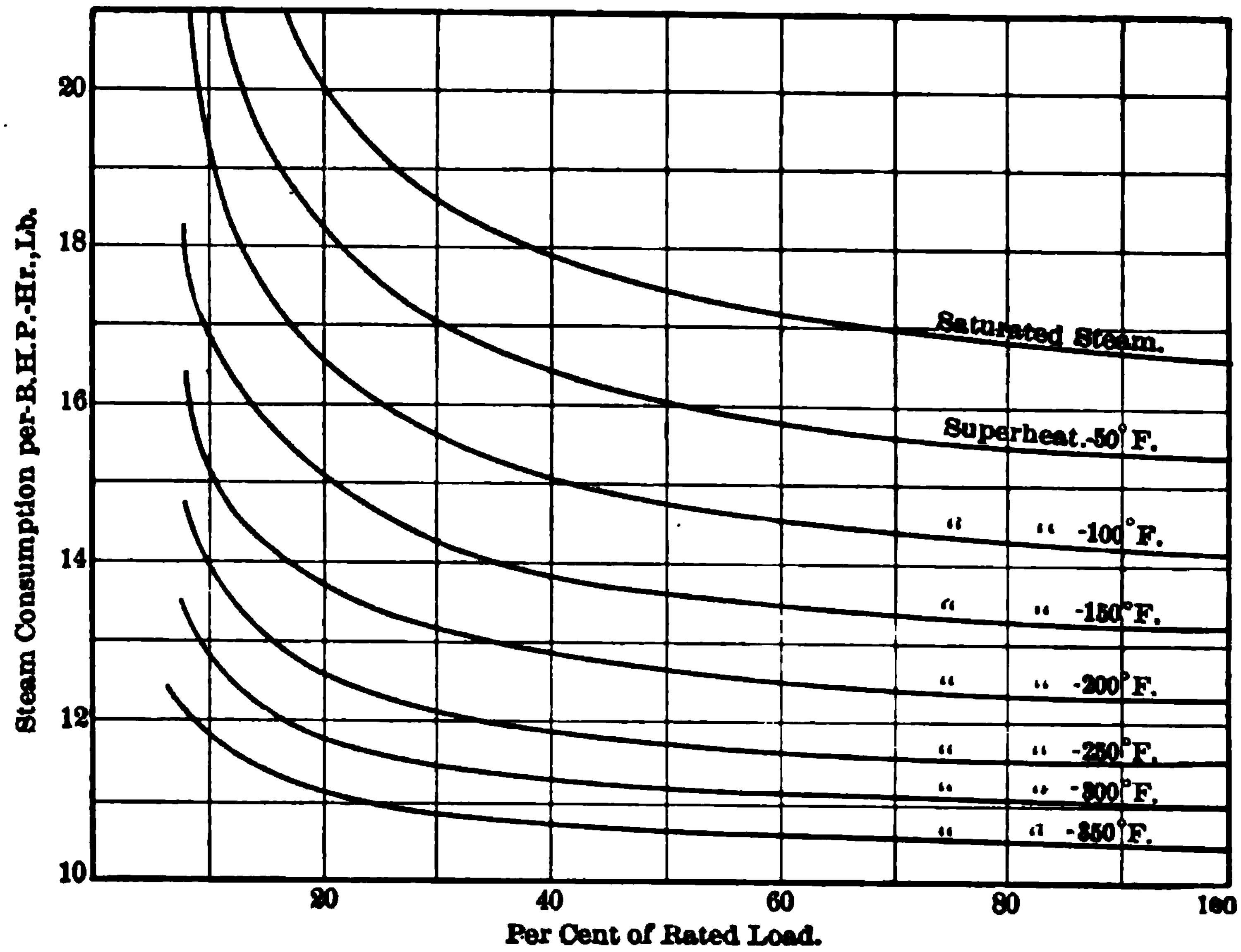


FIG. 160. Effect of Superheat on Steam Consumption.

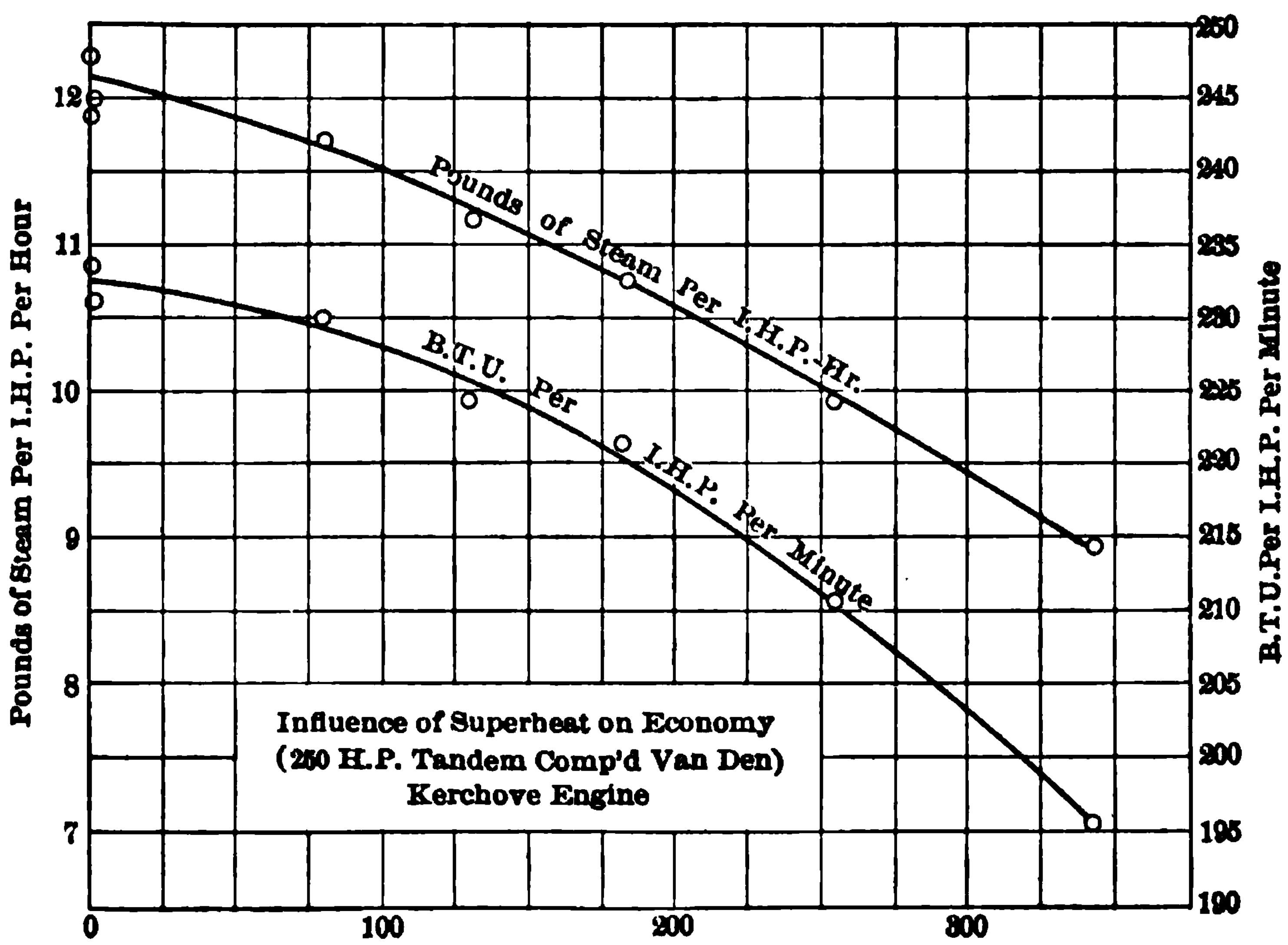


FIG. 161.

In the binary-vapor engine the working range has been considerably increased by substituting a highly volatile liquid, as sulphur dioxide, for the water which is ordinarily used as the cooling medium in the surface condenser.

The SO_2 in condensing the exhaust steam is itself vaporized and the vapor, under a pressure of about 175 pounds per square inch, used expansively in a secondary reciprocating engine. The exhausted SO_2 is discharged into a surface condenser in which it is liquefied by cooling water much the same as in refrigerating practice and used over and

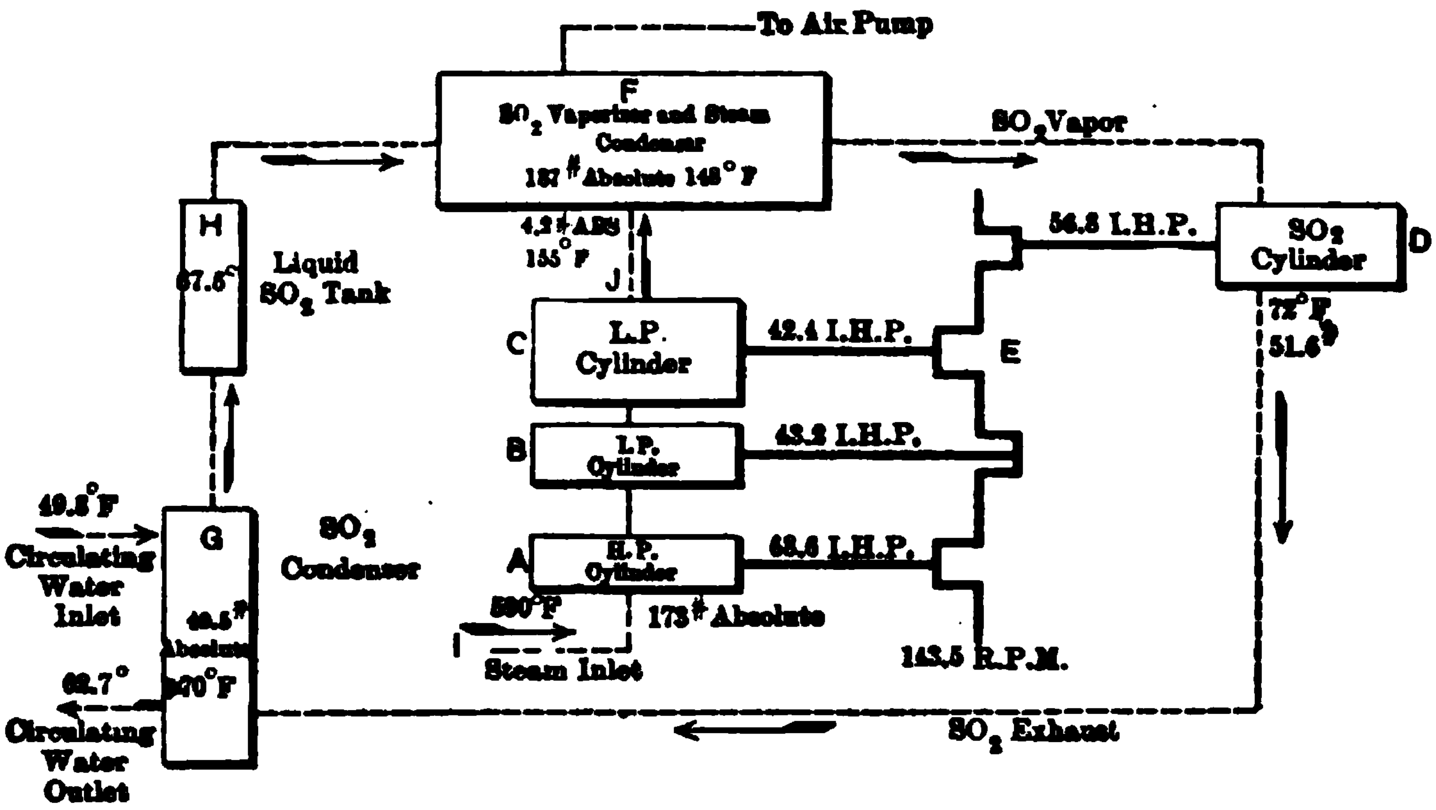


FIG. 162. Diagram of Binary-Vapor Engine.

over again. Referring to Fig. 162, which illustrates diagrammatically a binary-vapor engine at the Royal Technical High School, Berlin: *A*, *B*, and *C* are the three steam cylinders of an ordinary triple expansion engine and *D* the SO_2 cylinder. All four cylinders drive a common crank shaft *E*. *F* is a high-pressure surface condenser which acts as a vaporizer for the SO_2 and a condenser for the steam. *G* is a surface condenser which serves to condense the SO_2 vapor. *H* is a liquid SO_2 tank. The operation is as follows: Highly superheated steam enters the high-pressure steam cylinder at *I* and leaves the low-pressure cylinder at *J*, just as in any steam engine. The exhaust steam enters chamber *F* and is condensed by the liquid SO_2 passing through the coils. The condensed steam and entrained air are removed from the chamber by a suitable air pump. The steam in condensing gives up its latent heat to the liquid SO_2 and causes it to vaporize. The SO_2 vapor passes from the coils in chamber *F* to the SO_2 engine *D* and performs work. The exhausted SO_2 vapor flows from cylinder *D* to chamber *G*, and is condensed by cooling water flowing through a series

of tubes. The liquid SO_2 is collected in liquid tank *H* and thence is pumped into the coils in vaporizer *F*. The approximate temperatures and pressures at different points of the cycle are indicated on the diagram.

A number of experiments made by Professor E. Josse in the laboratory of the Royal Technical High School of Berlin on an experimental plant of about 200 horse power gave some remarkable results. A few of the tests made with highly superheated steam gave the following average figures:

I.H.P. (steam end)	146.4
Steam consumption per I.H.P. hour	12.8
I.H.P. (SO_2 end)	52.7
Percentage of power of SO_2 engine	35.9
Steam consumption per I.H.P. hour of combined engine	9.43

When operating under the most satisfactory conditions a performance of 8.36 pounds of steam per I.H.P. hour was recorded, corresponding to a heat consumption of 158.3 B.T.U. per minute, which is the best recorded performance to date (1907) in the history of steam-engine economy.

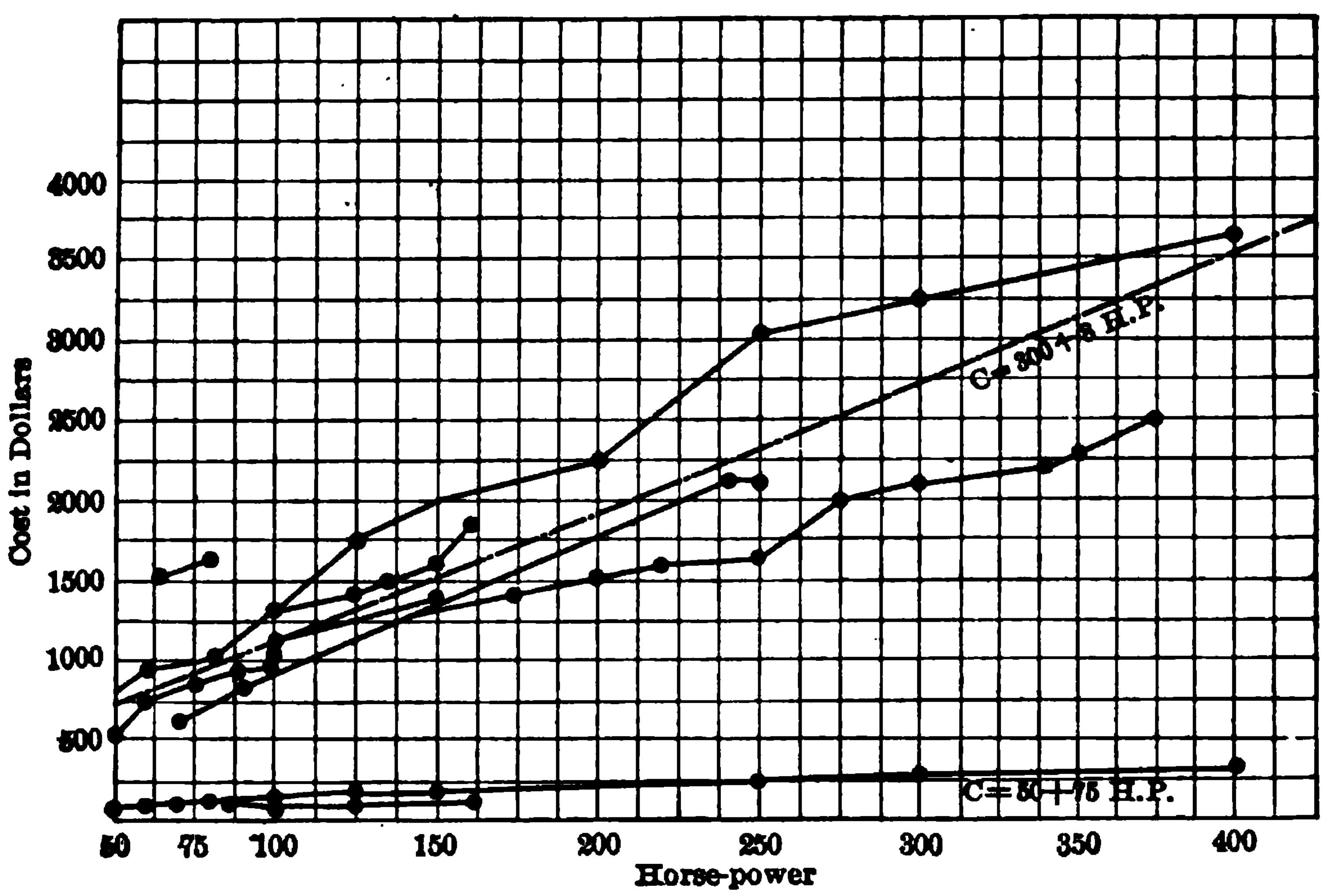


FIG. 163. Cost of Simple High-Speed Engines.

SO_2 does not attack the metal surface of the engine unless combined with water, in which case sulphurous acid is formed. There is, however, no danger from this cause, since the SO_2 being under greater pressure effectually prevents leakage of water into the SO_2 system.

The SO_2 cylinder requires no other lubrication than the SO_2 itself, which is of a greasy nature.

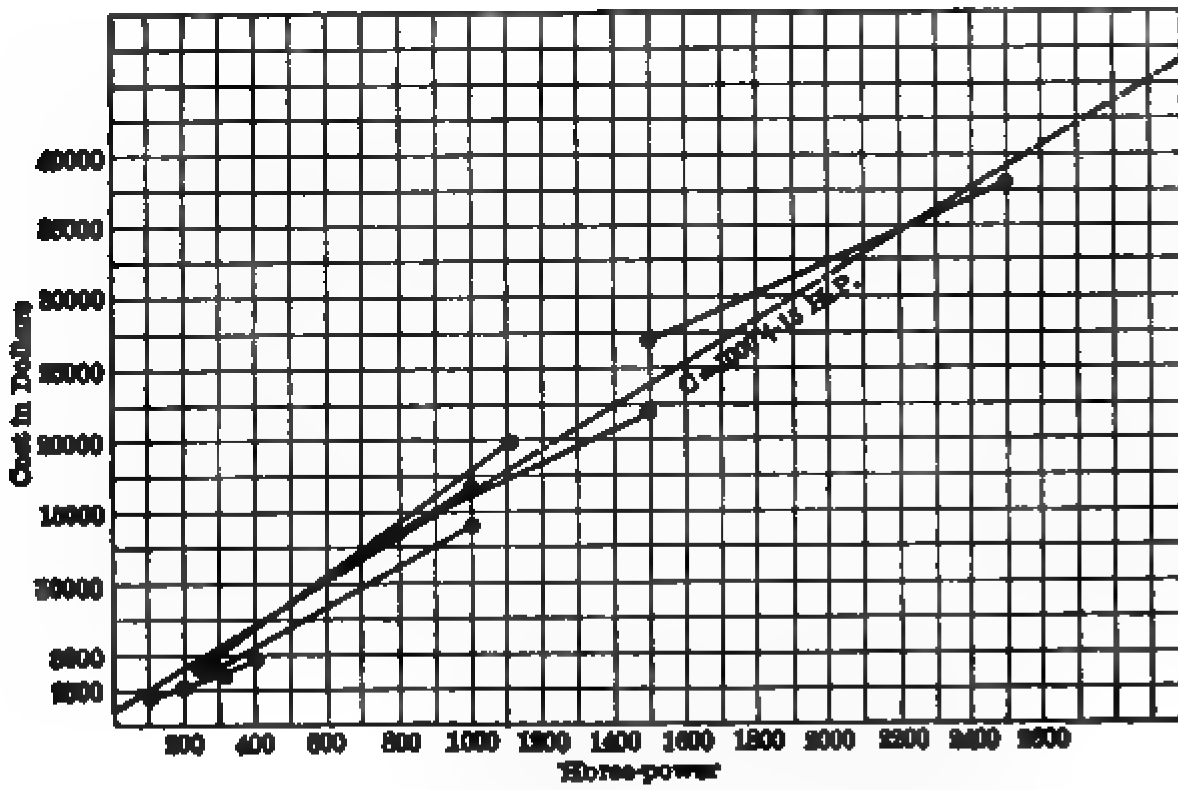


Fig. 164. Cost of High-Speed Compound Engines.

Cost in Dollars

Horse-power

Fig. 165. Cost of Low-Speed Engines, Simple and Compound.

Properties of SO_2: Trans. A.S.M.E., 25–181. *Binary-Vapor Engines:* Jour. Frank. Inst., June, 1903; Elec. World and Engr., Aug. 10, 1901; U.S. Cons. Reports, No. 1139, Sept. 14, 1901; Engr. U.S., Aug. 1, 1903; Sib. Jour. of Eng., March, 1902.

183. Cost of Engines. — In general the cost of engines per horse power diminishes as the size increases, but is of course governed by the style and workmanship. Average figures may be expressed as follows (*Engr.* U.S., Nov. 15, 1902, p. 750):

Simple high-speed engines.........	Cost in dollars = 300 + 8 × horse power
Setting, high-speed engines........	Cost in dollars = 60 + 0.75 × horse power
Compound high-speed engines......	Cost in dollars = 1000 + 15 × horse power
Simple low-speed engines..........	Cost in dollars = 1000 + 10 × horse power
Compound low-speed engines.......	Cost in dollars = 2000 + 13 × horse power
Setting, low-speed engines.........	Cost in dollars = 500 + 1.3 × horse power

These equations were deduced from the curves in Figs. 163 to 165, which were plotted from the actual costs of a large number of engines.

Rules for testing steam engines. — See Appendix C.

CHAPTER X.

STEAM TURBINES.

184. Classification. — The following outline gives a classification of a few well-known steam turbines:

Steam Turbines.	Impulse Type.	Single Stage.	De Laval.	Nozzle Expansion.
			Sturtevant.	
		Multi-Stage.	Curtis.	
			Kerr.	
			Terry.	
			Rateau.	
			Hamilton-Holzworth.	Blade Expansion.
	Reaction Type.	Multi-Stage.	Parsons.	
			Schulz.	
			Allis-Chalmers.	

In the impulse type the steam is expanded by suitable means before doing useful work; that is, its potential energy is first converted into kinetic energy. In the reaction type the conversion is not complete, the expansion taking place partly before doing work upon the wheel and partly within the blades of the wheel itself. Thus the steam gives up a portion of its energy by direct impulse in impinging against the blades or buckets and the balance by reaction in leaving them. The impulse type may be either single- or multi-stage, depending upon the number of divisions in which expansion takes place, but the reaction type is always multi-stage. The single-stage impulse machine has one row of buckets or vanes mounted on the periphery of a revolving disk and one set of stationary nozzles. The peripheral velocity is very high, ranging in practice from 700 to 1400 feet per second. In the multi-stage impulse machines the expansion is divided between a number of stages, each one exhausting through suitably porportioned nozzles into the next succeeding stage. The steam velocity is thereby very much reduced, and the peripheral velocity may be considerably lower for good efficiency, ranging in practice from 200 to 600 feet per second. In the reaction type the steam flows through a large number of rows of blades alternately fixed and revolving.

184a. General Elementary Theory. — A given weight of steam at a given pressure and temperature occupies a certain known volume and contains a known amount of heat energy. If the steam is permitted to expand to a lower pressure without receiving additional heat or giving up heat to surrounding bodies it is capable of doing a certain amount of work which will be the same whether the expansion takes place in the cylinder of a reciprocating piston engine, a rotary piston engine or the nozzles and blades of a steam turbine.

Let W = weight of steam, lbs. per sec.
E = energy given up by 1 pound of steam, ft.-lbs.
P_1 = initial pressure, lbs. per sq. in. abs.
P_n = final pressure, lbs. per sq. in. abs.
H_1 = initial heat contents per lb., B.T.U.
H_n = final heat contents per lb., B.T.U.

Then the heat available for doing useful work is

$$W\,(H_1 - H_n) \text{ B.T.U.}$$

If the steam expands against a resistance, as, for example, the piston of a reciprocating engine, the energy given up in forcing the piston forward may be expressed

$$E_1 = 778\,W\,(H_1 - H_n) \text{ ft.-lbs.} \qquad (72)$$

If the steam expands within a perfect nozzle the energy will be given up in imparting velocity to the steam itself, thus:

$$E_2 = W\frac{V_1^2}{2g} \text{ ft.-lbs.,} \qquad (72a)$$

in which V_1 = velocity of the jet in feet per second.

If the velocity of the jet is retarded to V_n feet per second, as by placing a series of vanes in its path, then the energy given up to the vanes (neglecting all losses) is

$$E = W\frac{V_1^2 - V_n^2}{2g}. \qquad (72b)$$

If the jet is brought to rest by the vanes (neglecting all losses), then $V_n = 0$ and the energy given up is

$$E_3 = W\frac{V_1^2}{2g}. \qquad (72c)$$

But $E_1 = E_2$. Hence,

$$778\,W\,(H_1 - H_n) = W\frac{V_1^2}{2g},$$

from which

$$V_1 = 223.9\sqrt{H_1 - H_n}.* \qquad (73)$$

* For most purposes it is sufficiently accurate to make 223.9 = 224.

The jet issuing from the nozzle is capable of exerting an *impulse* equal to F upon any object in its path, thus:

$$F = \frac{WV_1}{g} \text{ lbs.} \tag{74}$$

If A = the area of cross section of the jet in sq. ft. and γ = weight of steam, lbs. per cubic foot, then $W = \gamma AV_1$, or

$$F = \frac{\gamma AV_1^2}{g} \text{ lbs.} \tag{74a}$$

The *reaction*, *R*, of the jet against the nozzles is equal in value and opposite in direction to the impulse, or

$$R = F = \frac{WV_1}{g} = \frac{\gamma AV_1^2}{g}. \tag{74b}$$

The theoretical horse power developed by a jet of steam flowing at the rate of one pound per second may be expressed

$$\text{H.P.} = \frac{E}{550} = \frac{V_1^2 - V_n^2}{2g \times 550}, \tag{75}$$

in which V_1 = initial velocity of the jet, ft. per sec.

V_n = final velocity of the jet, ft. per sec.

Steam consumption per horse power hour:

$$W = \frac{3600}{\text{H.P.}}. \tag{75a}$$

Heat consumption, B.T.U. per horse power, per min.:

$$= \frac{W(H_1 - q_n)}{60}. \tag{75b}$$

in which q_n = heat of the liquid at pressure P_n.

Impulse efficiency of the jet = equation (72b) ÷ equation (72c).

$$E_i = \frac{V_1^2 - V_n^2}{V_1^2}. \tag{76}$$

Thermal efficiency:

$$E_t = \frac{H_1 - H_n}{H_1 - q_n}. \quad \text{See equation (68).} \tag{76a}$$

Efficiency ratio or " kinetic " efficiency:

$$E_r = \frac{2545}{W(H_1 - H_n)}. \quad \text{See equation (71).} \tag{76b}$$

Equations (72) to (76b) are general and are applicable to all turbines of whatever make.

The more important types of turbines will be discussed separately and an application of above equations will be made in each specific case.

FIG. 186. Section through a De Laval Turbine and Gear Casing.

185. The De Laval Turbine. — Fig. 166 shows a section through a De Laval steam turbine and gear case and illustrates the principles of the single-stage " velocity " type. The turbine proper, to the right of the figure, consists of a high-carbon steel disk *W* fitted at the periphery with a single row of drop-forged steel blades and inclosed in a cast-steel casing. The disk is secured to a light flexible shaft and is of such a cross section that the radial and tangential stresses throughout its mass are of constant value. A flexible shaft is employed which allows the wheel to assume its proper center of rotation and thus to operate like a truly balanced rotating body.* The shaft is supported by three bearings, *F*, *P*, and *N*. *N* is self-aligning and carries the greater part of the weight of the disk. *P* is a flexible bearing, entirely free to oscillate with the shaft, and its only function is to seal the wheel casing against leakage. The power is transmitted through a steel helical pinion *K* mounted on the extension of the turbine shaft *X*, to two large gears *E*, *E* at a reduction in speed of about 10 to 1. The blades, Fig. 167, are made with a bulb shank and fitted in slots milled

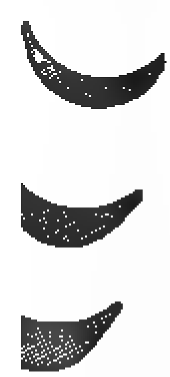


Fig 167. De Laval Blades.

in the rim of the wheel. The flanges, at the outer end of the blades, are brought in contact with each other and calked so as to form a continuous ring. The inlet and outlet angles of the blades are made alike and are 32 degrees for smaller sizes and 36 degrees for larger sizes.

The operation is as follows: Steam enters the steam chest *D*, Figs. 166 and 168, through the governor (shown in detail in Fig. 169) and is distributed to the various adjustable nozzles, varying in number from 1 to 15 according to the size of turbine. In the earlier types the nozzles were uniformly distributed around the circumference, but in the later types are arranged in groups. As illustrated in Fig. 168,

* The shaft diameter for a 100-H.P. turbine is but 1 inch and for a 300-H.P. approximately $1\frac{1}{1}\frac{3}{6}$ inches.

the nozzles are placed at an angle of 20 degrees with the plane of the disk. The steam is expanded adiabatically in the nozzles to the existing back pressure before it impinges at high velocity against the

Fig 168. De Laval Nozzle.

blades. After giving up its energy the steam passes into chamber *G*, Fig. 166, and out through the exhaust opening. Fig. 169 gives the details of the governor and vacuum valves. Two weights *B* are

Fig. 169. De Laval Governor.

pivoted on knife edges *A* with hardened pins *C* bearing on the spring *D*. *E* is the governor body, fitted in the end of the gear wheel shaft *K*, and has seats milled for the knife edges *A*. The spring seat *D* is

held against pins A by spiral concentric springs, the tension on which is adjusted by a milled nut I. When the speed exceeds the normal, centrifugal force causes the weights to fly outward and overcome the resistance of the springs. This pushes pin G against bell crank L, which in turn closes the double-seated valve, thus throttling the supply of steam. To prevent racing in case the load is suddenly removed the vacuum valve T is added to the governor mechanism. Its operation is as follows: The governor pin G actuates the plunger H under normal conditions without moving the plunger relative to the bell crank. In case the load is suddenly removed, centrifugal force pushes pin G against bell crank L until it reaches its extreme position and the valve is nearly closed and little steam enters the turbine. If this does not check the speed, plunger G overcomes the resistance of spring M, and H moves relative to L, and its adjustable projection O presses against valve stem T and allows air to rush into the turbine chamber through passage P.

The power of the turbine depends upon the number of nozzles in action, and these can be opened or closed by a hand wheel on each. Each nozzle performs its function as perfectly when operating alone as when operating in conjunction with others.

De Laval turbines are made in sizes ranging from 1½ to 300 horse power, condensing and non-condensing, and are designed to regulate within an extreme variation of 2 per cent from no load to full load.

The speeds vary from 10,600 r.p.m. for the largest size to 30,000 r.p.m. for the smallest, the gearing reducing these to 900 and 3000 r.p.m., respectively, at the shaft.

The diameter of the wheel varies from 4 inches in the smallest turbine to 30 inches in the largest, thus giving peripheral velocities of from 520 to 1310 feet per second.

De Laval Turbine: Prac. Engr., Jan. 1, 1910; Trans. A.S.M.E., 25–1056; Elec. World, July 29, 1905, p. 194, Oct. 26, 1901, p. 693; Eng. Rec., Oct. 19, 1901, p. 371; West. Elecn., June 4, 1904, p. 463; Machinery, Oct., 1904, p. 6, Nov., 1904, p. 123; Electrician, March 4, 1904; Power, Oct., 1905, p. 593.

See Table 48 for results of tests of turbines of this type.

186. Elementary Theory.— De Laval Turbine.— The maximum theoretical power developed by a jet of steam flowing through a nozzle is dependent only upon the *weight* of steam flowing per unit of time and the *initial velocity*. Therefore the higher the initial velocity for a given rate of flow the greater will be the power developed and the higher the efficiency.

* In Europe De Laval turbines are made as large as 3750 H.P. (See Power and Engr., May 10, 1910, p. 708.)

The maximum *weight* of steam discharged through a nozzle of any shape and for a given initial pressure is determined by the *area* of the narrowest cross section or *throat.*

To obtain the maximum *velocity* at the exit or *mouth,* for a given rate of flow, the nozzle should be proportioned so that expansion to the external pressure into which the nozzle delivers shall take place within the nozzle itself. If expansion in the nozzle is incomplete, sound waves will be produced and there will be irregular action and loss of energy. On the other hand, if expansion in the nozzle is carried below that of the external pressure at the mouth, sound waves will be produced with subsequent loss of energy even greater than in the former case.

Experimental and mathematical investigations indicate that the pressure at the narrowest section of an orifice or the throat of a nozzle through which steam is flowing falls to approximately 0.58 of the initial absolute pressure (with resultant velocity of about 1400 to 1500 feet per second) and any farther fall in pressure must take place beyond the narrowest section. Thus for back pressures greater than 0.58 of the initial (conveniently takes as $\frac{3}{5}$), maximum exit velocity may be ob-

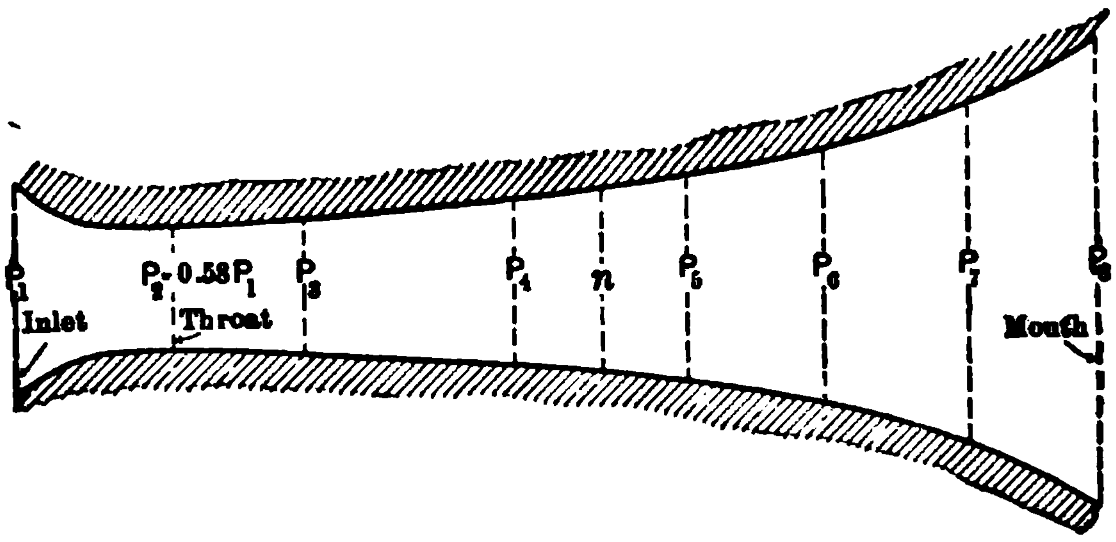


Fig. 170. Theoretically Proportioned Expanding Nozzle.

tained from orifices or nozzles of uniform cross section or with sides *convergent.* For back pressure less than 0.58 of the initial the nozzle must first *converge* from inlet to throat and then *diverge* from throat to mouth in order to obtain maximum velocity. Without the divergent portion of the nozzle the jet will begin to spread after passing the throat, and its energy will be given up in directions other than that of the original jet.

Fig. 170 shows a section through a theoretically proportioned expanding nozzle. The cross section of the tube at any point n may be calculated by means of equation

$$A_n = \frac{WS_n}{V_n}, \tag{76c}$$

in which

A_n = area in square feet.

W = maximum weight of steam discharged, lbs. per sec.

S_n = specific volume of the steam at pressure P_n.

For saturated steam $S_n = x_n u_n$,

in which x_n = quality of steam at pressure P_n after adiabatic expansion from pressure P_1.

u_n = specific volume of saturated steam at pressure P_n.

For superheated steam, see equation at bottom of page 131.

V_n = velocity of the jet, feet per second.

V_n may be determined from equation (73):

$$V_n = 223.9 \sqrt{H_1 - H_n}.$$

By substituting H_n = heat contents corresponding to pressure $P_n = 0.58\ P_1$ in equation (73) and (76c) the area at the throat may be readily determined. The cross-sectional area for other points in the tube may be determined in a similar manner by assigning values of H_n corresponding to the various pressures.

In case of a perfect nozzle $H_1 - H_n$ represents the heat given up toward producing velocity by adiabatic expansion from pressure P_1 to P_n. In the actual nozzle the frictional resistance of the tube serves to increase its dryness fraction, but in doing so it decreases the amount of energy the steam is capable of giving up towards increasing its own velocity. If y one-hundredths of the heat $H_1 - H_n$ is utilized in overcoming frictional resistance, then the resulting velocity will be

$$V = 223.9 \sqrt{(1 - y)(H_1 - H_n)}. \tag{76d}$$

The quality of the steam after expanding to P_n against the resistance will be higher by an amount

$$I_n = \text{increase in quality} = \frac{y\,(H_1 - H_n)}{r_n}, \tag{76e}$$

in which r_n = heat of vaporization at pressure P_n.

The curves in Fig. 171, calculated by means of equations (76b) and (73), show the relationship between velocity, quality, pressure and kinetic energy for all points in a theoretically perfect nozzle expanding one pound of dry steam per second from an initial absolute pressure of 190 pounds to a condenser pressure of one pound.

The curves in Fig. 172 are based upon the experiments of Gutermuth

(*Zeit. d. Ner. Ingr.*, Jan. 16, 1904) and show the effect of a few shapes of nozzles and orifices on the actual weight of steam discharged for various rates of initial and final pressures, the smallest section of the tube remaining constant.

The nozzles of most commercial types of steam turbines are made with straight sides as in Fig. 168, so that only the area at the mouth need

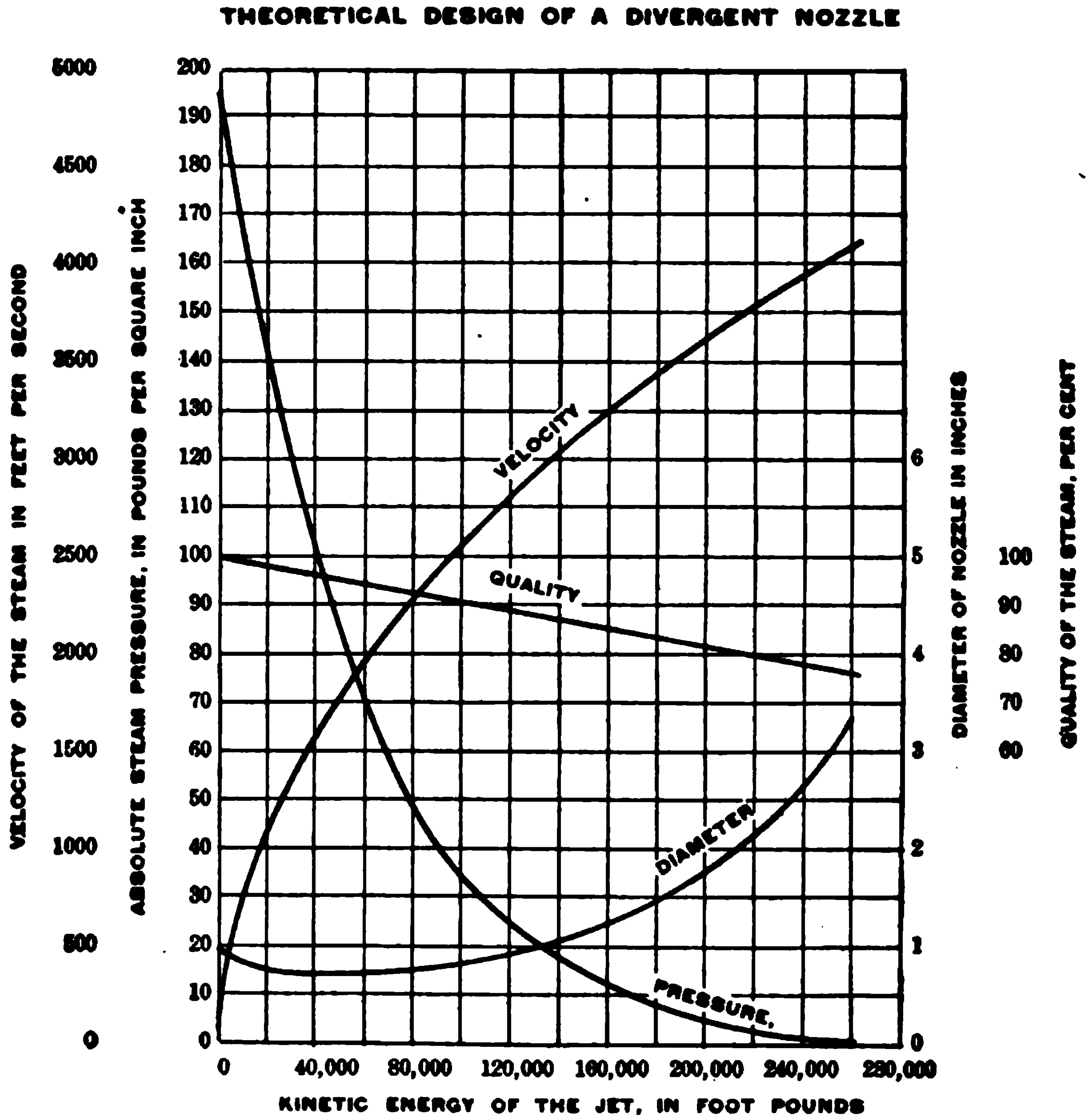


FIG. 171.

be determined in addition to that at the throat in order to lay out the shape of the tube.

Equations (73) and (76b) are general and are applicable to steam of any quality, wet, dry, or superheated. For steam initially dry and saturated *Napier's* rule offers a simple means of determining the area at the throat, thus:

$$W = \frac{A_0 P_1}{70} \text{ for } P_n = \text{ or } < \frac{3}{5} P_1. \qquad (76e)$$

$$W = 0.029\, A_0 \sqrt{P_n (P_1 - P_n)} \quad \text{for} \quad P_n > \tfrac{3}{5} P_1,$$

in which

W = maximum weight of steam discharged, lbs. per sec.
A_0 = area at the throat, sq. in.
P_1 = absolute initial pressure, lbs. per sq. in.
P_n = absolute back pressure, lbs. per sq. in.

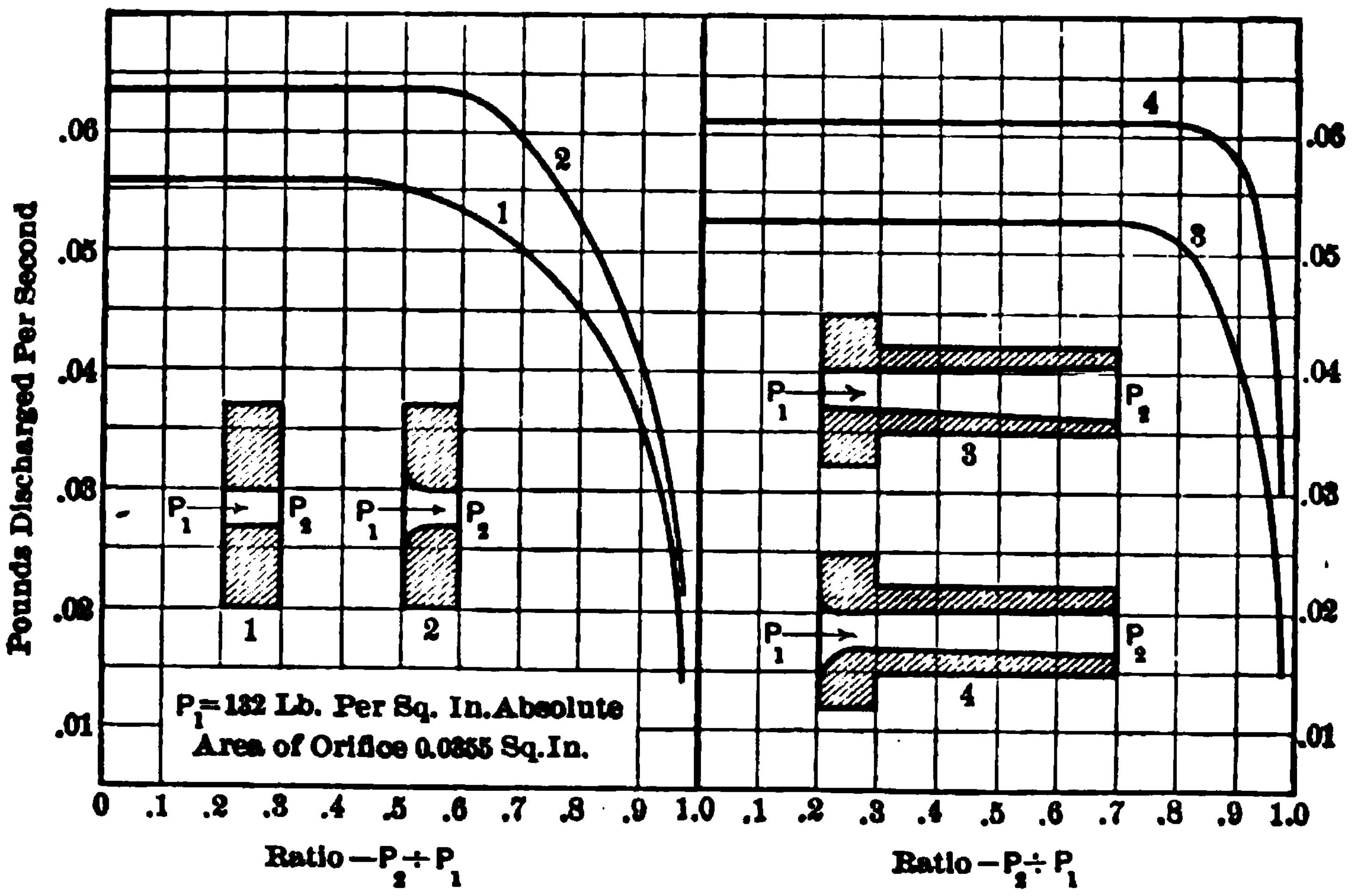


FIG. 172. Flow of Steam through Nozzles.

Moyer ("The Steam Turbine," 1st Edition, p. 40) states that the ratio of the area of a correctly proportioned nozzle at the throat A_0 to the area at any point A_n is very nearly proportional to the ratio of the pressure at point A_n to the initial pressure, or

$$\frac{A_0}{A_n} = \frac{P_1}{P_n}. \tag{76f}$$

The entrance to the tube is rounded by any convenient curve.

The length of the tube may be roughly approximated by the following formula:

$$L = \sqrt{15\, A_0}, \tag{76g}$$

in which

L = length between the throat and mouth, in inches.
A_0 = area at the throat, sq. in.

Practice shows that the cross section of a nozzle, whether circular, elliptical, square or rectangular (the latter with rounded corners), has very little influence on the efficiency provided the inner surfaces are smooth and the ratio of the area at the throat to that of the mouth is

correctly proportioned. The *velocity* efficiency of a properly proportioned nozzle with straight sides is about 95 to 97 per cent, corresponding to an *energy* efficiency of 92 to 94 per cent, so that it is not considered worth while to attempt to follow the more difficult exact curves.

Example: — Find the smallest cross section of a frictionless conically divergent nozzle for expanding one pound of steam per second from an absolute initial pressure of 190 pounds to an absolute back pressure of 2 pounds and find six intermediate cross sections where the pressures will be 70, 30, 14.7, 8, 4 and 2 lbs. respectively. Compare the velocity and energy of the jet issuing from this nozzle with those of an actual nozzle in which 10 per cent of the heat energy is lost in friction.

From steam and entropy tables we find the values of H, x, u, for absolute pressures corresponding to 190, 0.58 × 190 = 110, 70, 30, etc., lbs. per square inch as follows (theoretical nozzle):

	H.	x.	u.	$S = xu$.
P_1 = 190	1197.3	1.00	2.405	2.394
P_2 = 110*	1152.6	0.960	4.047	3.878
P_3 = 70	1117.9	0.932	6.199	5.775
P_4 = 30	1057.2	0.887	13.75	12.27
P_5 = 14.7	1011.3	0.857	26.78	22.95
P_6 = 8	947.8	0.834	47.26	39.29
P_7 = 4	935.6	0.810	90.4	73.2
P_8 = 2	899.3	0.788	173.1	137.0

* $P_2 = 0.58\ P_1$ (= pressure at throat).

If entropy tables or charts are not available, values H_1 to H_8 and x_1 to x_8 may be determined as outlined in equations (66b) to (67g).

The different quantities for the theoretical nozzle will be calculated for the exit pressure $P_n = P_8 = 2$ lbs. per sq. in absolute.

$$V_8 = 223.9\sqrt{H_1 - H_8}$$
$$= 223.9\sqrt{1197.3 - 899.3}$$
$$= 3865 \text{ feet per second.}$$

$$E_8 = 778\ (H_1 - H_8)$$
$$= 778\ (1197.3 - 899.3)$$
$$= 232{,}000 \text{ foot-pounds.}$$

$$A_8 = \frac{WS}{V}$$
$$= \frac{1 \times 137}{3865}$$
$$= .0353 \text{ square foot.}$$

$$d_8 = \sqrt{\left(\frac{144 \times 4}{\pi}\right) A} = 13.56 \sqrt{A}$$
$$= 13.56 \sqrt{.0353}$$
$$= 2.54 \text{ inches.}$$

$$F_8 = \frac{W V_8}{g}$$
$$= \frac{3865}{32.2}$$
$$= 120 \text{ pounds.}$$

THEORETICAL NOZZLE.

Quantity		V Ft. per Sec.	E Ft.-Lbs.	A Sq. Ft.	d Inches.	F Pounds.
Formula		(73)	(72)	(76c)		(74)
Pressures	110	1,496	34,767	.00259	0.693	46.4
	70	1,995	61,853	.00269	0.702	61.98
	30	2,650	107,485	.00461	0.919	82.3
	14.7	3,053	144,742	.00745	1.1	94.8
	8	3,339	173,207	.0119	1.46	103.7
	4	3,624	203,968	.0202	1.92	112.5
	2	3,865	232,000	.0353	2.54	120.0

In the actual nozzle these values will be modified because of the frictional losses. Thus for $P_n = 2$ lbs.,

$$V_8 = 223.9 \sqrt{(1 - y)(H_1 - H_8)}$$
$$= 223.9 \sqrt{(1 - 0.1)(1197.3 - 899.3)}$$
$$= 3667 \text{ ft. per sec.}$$

$$E_8 = 778 (1 - 0.1)(1197.3 - 899.3) = 208{,}800 \text{ ft.-lbs.}$$

$$x_8' = x_8 + I_8 = x_8 + \frac{y (H_1 - H_8)}{q_8}$$
$$= 0.788 + \frac{0.1 (1197.3 - 899.3)}{1021}$$
$$= 0.788 + 0.029$$
$$= 0.817.$$

$$A_8 = \frac{W x_8' u_8}{V_8}$$
$$= \frac{0.817 \times 173.1}{3667}$$
$$= 0.0386 \text{ sq. ft.,}$$

from which

$$d_8 = 2.66 \text{ inches.}$$

$$F_8 = \frac{WV_8}{g} = \frac{3668}{32.2} = 114 \text{ pounds.}$$

These various factors for all given pressures have been calculated in a similar manner and are as follows:

ACTUAL NOZZLE.

Quantities		V Ft. per Sec.	E Ft.-Lbs.	x'.	A Sq. Ft.	d Inches.	F Ft.-Lbs.
Pressures	110	1,420	31,317	.9658	.00275	0.711	44.1
	70	1,893	55,632	.9414	.00286	0.723	58.8
	30	2,515	98,257	.9026	.00493	0.951	78.12
	14.7	2,894	130,050	.876	.0080	1.2	98.8
	8	3,168	155,858	.856	.0127	1.53	98.4
	4	3,438	183,581	.836	.0220	2.01	106.8
	2	3,667	208,800	.817	.0386	2.66	114.0

Many of these values may be determined directly from the *Mollier* or total heat-entropy diagram as described in Appendix H; in fact, the Mollier diagram has to all intents and purposes supplanted the steam tables in this connection. For superheated steam the diagram is extremely useful in avoiding laborious calculations.

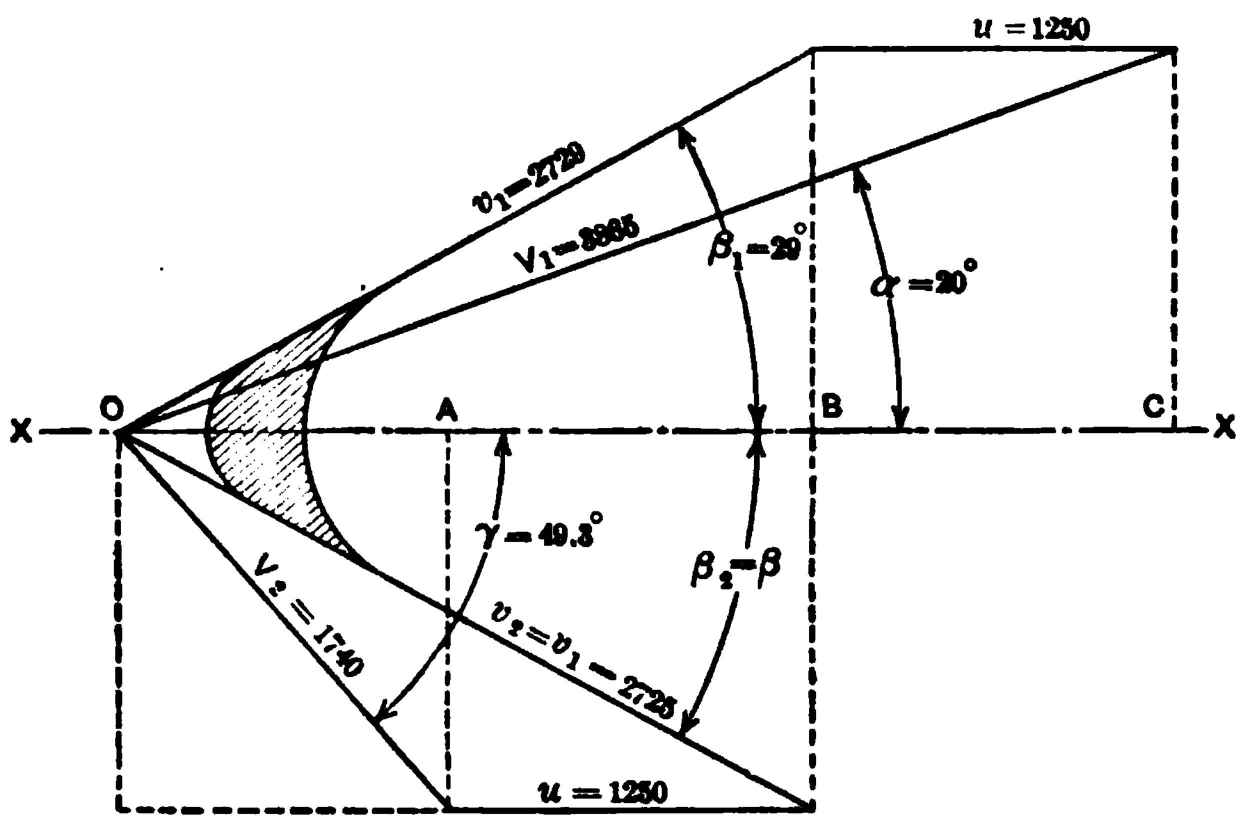


Fig. 172a. Velocity Diagram. Ideal Single-stage Impulse Turbine.

Fig. 172a gives a diagrammatic arrangement of the blades in a De Laval turbine. The nozzle directs the steam against the blades with *absolute* velocity V_1 and at an angle α with the plane of the wheel XX. Since the wheel is moving at a velocity of u feet per second, the velocity

v_1 of the steam *relative* to the wheel is the resultant of V_1 and u. The angle β_1 between v_1 and XX will be the proper blade angle at entrance. If the blade curve makes this angle with the direction of motion of the wheel no shock will be experienced when the steam enters the blades. For convenience in construction the exit angle β_2 is made the same as the entrance angle β_1. Neglecting frictional losses in the blade channels the *relative* exit velocity will be $v_2 = v_1$, and the *absolute* velocity V_2 is the resultant of v_2 and u. The impulse exerted by the jet in striking the vanes is $\frac{W}{g} v_1$, and its component in the direction of motion is $\frac{W}{g} v_1 \cos \beta_1 = \frac{W}{g}(V_1 \cos \alpha - u)$. As the jet leaves the vanes the impulse is $-\frac{W}{g} v_2 \cos \beta_2 = -\frac{W}{g}(V_2 \cos \gamma + u)$.

The *total pressure* acting on the vanes, or the actual driving impulse, is

$$P = \frac{W}{g}\left\{ V_1 \cos \alpha - u - (-V_2 \cos \gamma + u)\right\}$$

$$= \frac{W}{g}(V_1 \cos \alpha + V_2 \cos \gamma). \tag{77}$$

Equation (77) may also be expressed

$$P = \frac{W}{g} \cdot 2\,(V_1 \cos \alpha - u). \tag{77a}$$

The resultant *axial force* or *end thrust* is

$$F = \frac{W}{g}(V_1 \sin \alpha - V_2 \sin \gamma). \tag{77b}$$

Evidently if $\alpha = \gamma$ and $V_1 = V_2$ there will be no end thrust, since $V_1 \sin \alpha - V_2 \sin \gamma$ will be zero.

The *work done* is

$$Pu = \frac{W}{g} u\,(V_1 \cos \alpha + V_2 \cos \gamma), \tag{77c}$$

or, using equation (77a) in place of (77),

$$Pu = \frac{W}{g} \cdot 2\,u\,(V_1 \cos \alpha - u)$$

$$= \frac{W}{g} \cdot 2\,(uV_1 \cos \alpha - u^2). \tag{77d}$$

By making the first derivative equal to zero

$$\frac{d}{du}\left\{\frac{W}{g} 2\,(uV_1 \cos \alpha - u^2)\right\} = V_1 \cos \alpha - 2\,u = 0,$$

or

$$u = \tfrac{1}{2}\,V_1 \cos \alpha.$$

That is, for *any nozzle angle* α the work done, Pu, has its greatest value when $u = \frac{1}{2} V_1 \cos \alpha$ or $\gamma = 90°$, whence

$$Pu = W \frac{V_1^2}{2g} \cos^2 \alpha. \tag{77e}$$

The work for *any initial velocity* V_1 becomes a maximum when $\alpha = 0$ and $u = \frac{1}{2} V_1$. This condition can only occur for a complete reversal of jet and zero final velocity. Substitute $\alpha = 0$ and $u = \frac{1}{2} V_1$ in equation (77d).

$$Pu = \frac{W V_1^2}{2g},$$ which is necessarily the same as equation (72c).

In the actual turbine the various velocities will be less than those as obtained on account of the frictional resistance in the blades, and the velocity diagram should be modified accordingly.

Example. Lay out the blades (theoretical and actual) for the nozzle in the preceding example, assuming that the jet impinges against the wheel at an angle of 20 degrees and that the peripheral velocity is 1250 feet per second.

Theoretical Case.

Lay off $V_1 = 3865$ feet per second in direction and amount as shown in Fig. 172a and combine it with $u = 1250$ feet per second; this gives v_1, the relative entrance velocity as 2725 feet per second and β, the entrance angle as 29 degrees.

Lay off $v_2 = v_1$ at an angle $\beta_2 = \beta_1$ and combine with u; this gives V_2, the *absolute* exit velocity, as 1740 feet per second.

The theoretical energy available for doing work is

$$E = \frac{W}{2g}(V_1^2 - V_2^2)$$
$$= \frac{1}{64.4}(3865^2 - 1740^2) = 185{,}000 \text{ ft.-lbs.}$$

The difference between 232,000 and 185,000 = 47,000 ft.-lbs. is evidently the kinetic energy lost in the exhaust due to the exit velocity.

The pressure exerted by the steam on the buckets is

$$P = \frac{W}{g}(V_1 \cos \alpha + V_2 \cos \gamma)$$
$$\doteq \frac{1}{32.2}(3865 \times 0.9397 + 1740 \times 0.65166)$$
$$= 148 \text{ pounds.}$$

The theoretical impulse efficiency is

$$\frac{V_1^2 - V_2^2}{V_1^2} = \frac{3865^2 - 1740^2}{3865^2} = 0.797$$

The theoretical horse power per pound of steam flowing per second is

$$\text{H.P.} = \frac{185{,}000}{550} = 336.$$

Theoretical steam consumption per H.P.-hr. is

$$\frac{3600}{336} = 10.7 \text{ pounds.}$$

Actual Case.

Proceed as in the theoretical case, using the actual absolute velocity $V_1 = 3865\sqrt{1-y} = 3865\sqrt{1-0.10} = 3667$ feet per second in place of the theoretical value $V_1 = 3870$. Lay off $V_1 = 3667$ at an angle of 20° as before and combine with $u = 1250$, Fig. 172b.

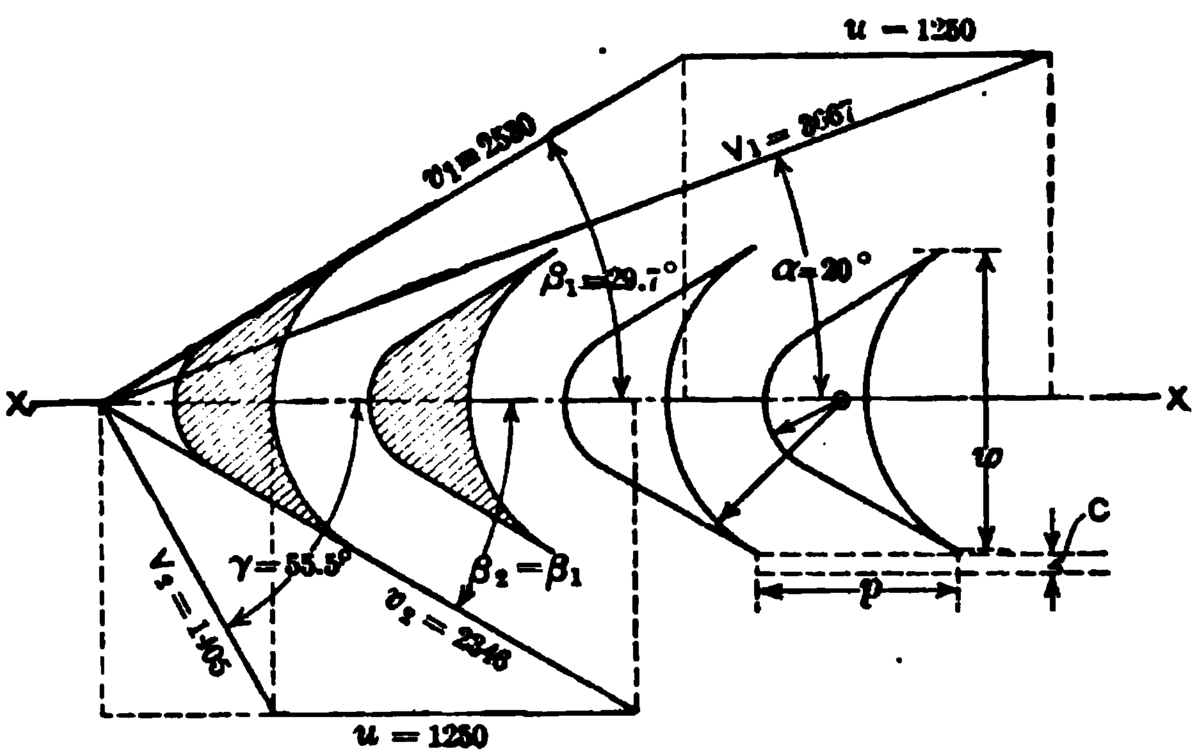


Fig. 172b. Velocity Diagram as Modified by Friction Losses.

The resultant $v_1 = 2530$ is the velocity of the jet relative to the wheel, and the entrance angle β is found to be 29.7 degrees. The relative exit velocity v_2 will be less than v_1 because of the blade friction.

Assume the loss of energy ϕ from this cause to be 14 per cent; then, since the velocity varies as the square root of the energy,

$$\begin{aligned} v_2 &= v_1\sqrt{1-\phi} \\ &= 2530\sqrt{1-0.14} \\ &= 2346 \text{ ft. per second.} \end{aligned}$$

The resulting absolute velocity V_2 is found from the diagram to be $V_2 = 1405$ ft. per second.

Since the loss of energy in the nozzle is

$$\frac{V_1^2 - (1-y)V_1^2}{2g},$$

and that in the blade

$$\frac{v_1^2 - (1-\phi)V_1^2}{2g},$$

the remaining energy, deducting both losses in the nozzle and the blades, is

$$\frac{W}{2g}(V_1^2 - yV_1^2 - \phi v_1^2 - V_2^2)$$

$$= \frac{1}{64.4}(3865^2 - 0.1 \times 3865^2 - 0.14 \times 2530^2 - 1405^2)$$

$$= 164{,}200.$$

The losses due to windage, leakage past the buckets and mechanical friction must be deducted from these figures to give the actual energy available for doing useful work. Assuming a loss of 15% due to this cause, the work delivered is

$$0.85 \times 164{,}200 = 139{,}570 \text{ ft.-lbs.}$$

The efficiency in the ideal case was found to be 0.797 and the available energy 185,000 ft.-lbs.

The efficiency, deducting the loss due to friction, etc., is

$$\frac{139{,}570}{185{,}000} \times 0.797 = 0.60.$$

The horse power delivered is

$$\frac{139{,}570}{550} = 254.$$

Steam consumption per horse-power hour is

$$\frac{3600}{254} = 14.2 \text{ pounds.}$$

The heat consumption, B.T.U. per H.P., per minute is

$$\frac{16.4\,(1197.3 - 94)}{60} = 316.$$

Assuming the revolutions per minute to be 10,000, the mean diameter of the wheel to give a peripheral velocity of 1250 ft. per second is

$$\frac{1250 \times 60}{1000 \times 3.14} = 2.39 \text{ ft., or } 28.6 \text{ inches.}$$

The determination of the height and width of vanes, clearance between nozzles and blades, etc., are beyond the scope of this work and the reader is referred to the accompanying bibliography.

Blade Design for De Laval Turbines: Moyer, Steam Turbine, Chap. IV; Power, Mar. 17, 1908, p. 391.

Flow of Steam through Nozzles: Jour. A.S.M.E. Mid. Nov., 1909, April, 1910, p. 537; Engineering, Feb. 2, 1906; Engr., Lond., Dec. 22, 1905; Eng. Rec., Oct. 26, 1901; Power, May, 1905; Eng. News, Sept. 19, 1905, p. 204; French, Steam Turbines, Chap. XI; Pro. Inst. Civ. Engrs., Feb. 2, 1906.

Design of Turbine Disks: Engr., Lond., Jan. 8, 1904, p. 34, May 13, 1904, p. 481; Jude, Theory of the Steam Turbine, Chap. XIII; Thomas, Steam Turbines, Chap. VI.

Steam Turbine Efficiency: Power, Feb., 1906, p. 83; Jude, Theory of the Steam Turbine, Chap. VIII.

Critical Velocity of Shafting: Jour. A.S.M.E., June, 1910, p. 1060; Power, Sept., 1903, p. 484; Stodola, Steam Turbines, p. 177; Jude, Theory of the Steam Turbine, Chap. XVI; French, Steam Turbines, Chap. XV.

Tests of De Laval Turbines: Eng. Rec., Aug. 2, 1902, p. 100; Am. Elecn., Aug., 1905, p. 445; Engr. U. S., Aug. 1, 1905, p. 526; Eng. and Min. Jour, Nov. 3, 1904, p. 706; Machinery, Aug., 1904, p. 560; Eng. News, June 19, 1905, p. 62.

187. Terry Turbine. — Fig. 173 shows a section through a Terry turbine, illustrating an application of the impulse type with two or more *velocity stages.* The rotor, a single wheel consisting of two steel

FIG. 173. Section through Terry Steam Turbine.

disks held together by bolts over a steel center, is fitted at its periphery with pressed steel buckets of semicircular cross section. The inner surface of the casing is fitted with a series of gun-metal reversing bucket arranged in groups, each group being supplied with a separate nozzle. The steam issuing from nozzle N, Fig. 174, strikes one of the buckets, B, on the wheel and, since the velocity of the buckets is comparatively low, is reversed in direction and directed into the first one of the reversing chambers. The chamber redirects the jet against the wheel, from which it is again deflected; this is repeated four or more times until the available energy has been absorbed by the rotor. Terry turbines are made in a number of sizes varying from 2 to 800 horse power, and operate

at speeds varying from 210 feet per second in the smaller machine to 260 feet per second in the larger. These low speed limits compared with the speed of single-stage De Laval turbines are made possible by the application of the velocity stage principle in the use of the reversing buckets. The rotor of the smaller machine is 12 inches in diameter and runs at 4000 r.p.m., and that of the larger, 48 inches, running at 1250

Fig. 174. Arrangement of Buckets and Reversing Chambers in a Terry Steam Turbine.

r.p.m. Since the flow of steam into and from the buckets is in the plane of the wheel there is no end thrust.

For a description of the Bliss, Dake, Sturtevant and Wilkinson steam turbines with results of tests see "Small Steam Turbines," by G. A. Orrok, Jour. A.S.M.E., May, 1909, and contributed discussion, Sept., 1909. See also, "The Development of the Small Steam Turbine," Eng. Mag., Dec., 1908, and Jan., 1909.

188. Kerr Turbine. — Fig. 175 shows a longitudinal section through, and Fig. 176 a sectional elevation of, a Kerr steam turbine. This turbine is of the impulse type and built on the principle of the Pelton water wheel, which it resembles in many respects. The rotor consists of a series of steel disks *R, R*, Fig. 175, mounted on a steel shaft. A series of drop-forged mild steel buckets of the double-cup type are secured to the periphery and riveted in dovetail slots. The stator is made up of a number of cast-iron diaphragms *S, S*, with circular rims, which are tongued and grooved and when drawn together form a continuous cylinder. Square cold-rolled steel nozzle bodies *N, N* are expanded and beaded in the diaphragm near the rim and the nozzles screwed into them. The bearings *B, B* are of the oil-ring type; no thrust blocks are necessary, as each element is practically balanced. The operation is as follows: Steam enters the turbine at inlet *A* and passes through balanced throttling valve *V* (controlled by governor *G*) to the circular cored space *H, H* extending around the

FIG. 175. Kerr Steam Turbine — Longitudinal Section.

entire casing. Space *H, H* acts as an equalizer and insures uniform admission to the first row of nozzles, where steam is partially expanded and the kinetic energy imparted to the rotor through the medium of the buckets. The steam leaves the buckets at practically zero velocity

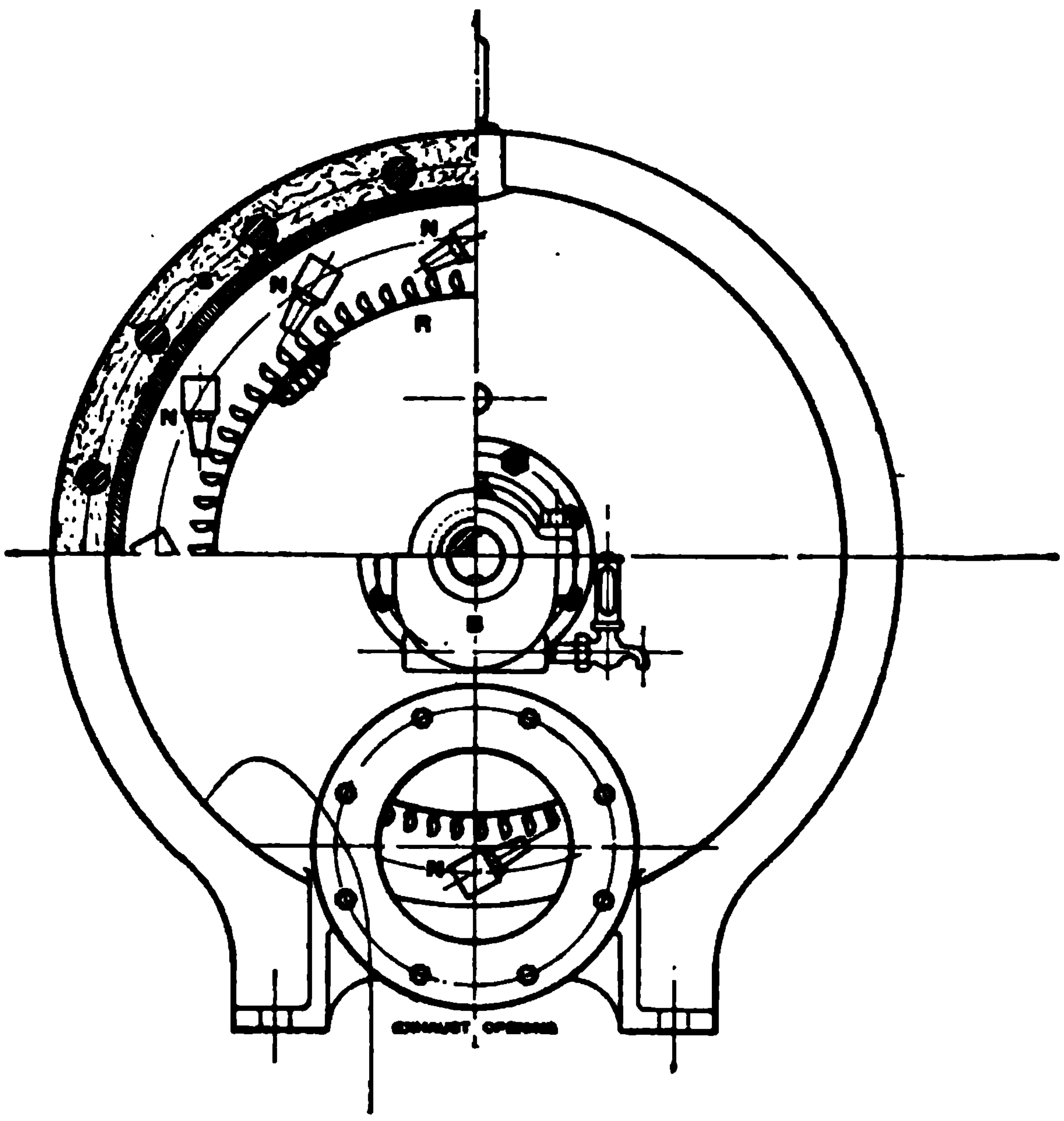


FIG. 176. Kerr Steam Turbine; Sectional End Elevation.

and is again expanded through the second set of nozzles. This process is repeated in each stage and the exhaust steam leaves the turbine at *O*. Fig. 177 shows a diagrammatic arrangement of the governor. The governor weight is turned from solid steel and split into two pieces

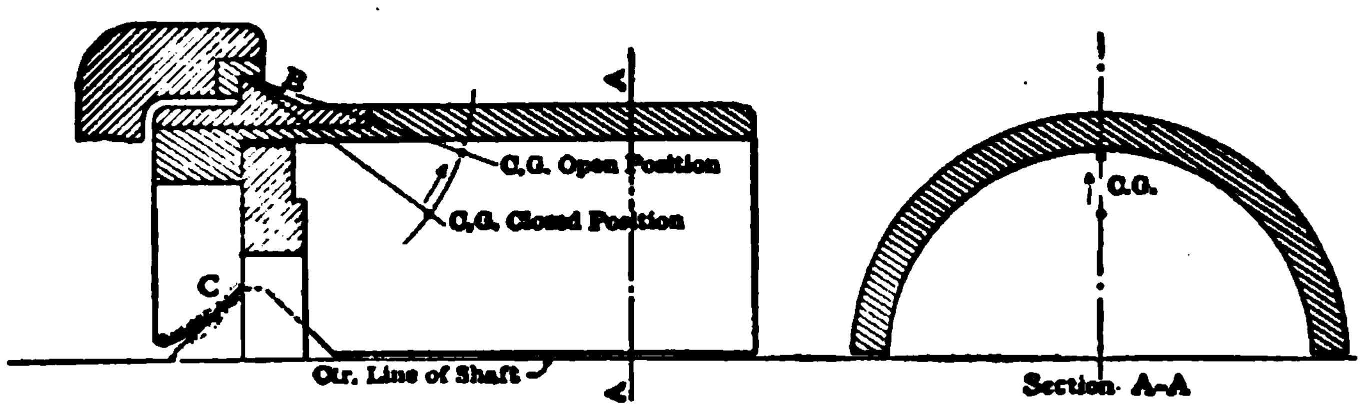


FIG. 177. Kerr Steam Turbine Governor.

of semi-cylindrical form with the center of gravity near the center of the shaft. The weights are supported at three points. The hardened steel knife edge at *B* is of sufficient length for the stresses involved. The curve of rolling contact *C* is such that the bearing between the weight

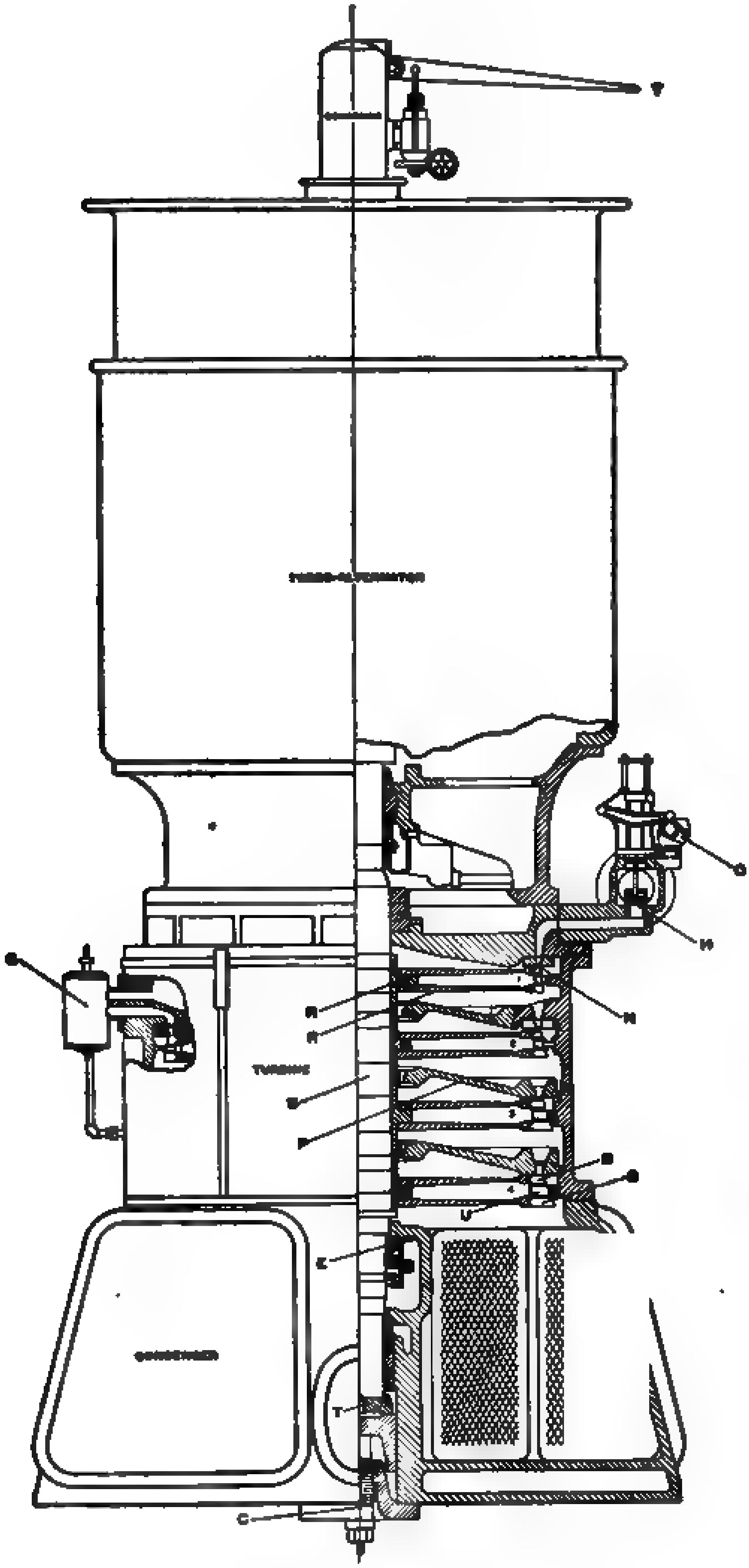


Fig. 178. Four-stage Vertical Curtis Turbo-Generator. Base Condenser Type.

and the cam collar is always on the line of centers. The outward movement of the weights compresses the spring and operates, through lever connections, the balanced piston valve controlling the flow of steam. The movement of the center of gravity is indicated.

The Kerr turbine is very simple in design, compact, noiseless, and low in cost of repairs. Its performance compares favorably with all other types of turbines of similar size and capacity.

An 18-inch Kerr turbine direct connected to a multi-stage Worthington centrifugal pump at the Armour Institute of Technology gives a steam consumption when running non-condensing comparable with that of high-grade non-condensing engines.

Rateau Turbine: Trans. A.S.M.E., 25–782; Eng. Mag., Oct., 1903, p. 49; St. Ry. Jour., April 18, 1903.

Zölly Steam Turbine: Elec. Rev., Sept. 2, 1904.

189. The Curtis Steam Turbine. — Figs. 178 to 183 show the general arrangement and a few details of the Curtis steam turbine, which is of the compound or multi-stage velocity type. The total expansion is carried out in one or more compartments or stages, each stage comprising a set of expanding nozzles and a wheel carrying two or more rows of buckets. A high initial velocity is given to the jet in each stage by expansion in the nozzles as in the De Laval, and the energy absorbed by successive action upon the series of moving and stationary vanes arranged somewhat as in the Parsons turbines, paragraph 192. In the latter, however, the difference in pressure between the two sides of each vane induces flow by continuous expansion, while in the former the moving vanes in any one stage simply absorb the kinetic energy already created by expansion in the nozzle. The action is as follows: Steam enters stage (1), Fig. 180, through the first set of nozzles, and is partially expanded. With the resulting initial velocity it impinges against the first row of moving blades and gives up part of its energy, and is then deflected through the adjoining stationary blades to the next set of moving vanes, where its velocity is still further reduced, and so on until it has been brought practically to rest. From this stage the steam flows at reduced pressure through nozzles of stage (2), which are sufficient in number and in size to afford the greater area required by the increased volume. In expanding in these nozzles it acquires new velocity and gives up energy to the moving blades as before. This process is repeated through two to five stages, depending upon the size of turbine. Fig. 178 shows a partial section of a four-stage 5000-kilowatt machine. *R*, *R* are sections through the revolving wheels, which in this particular turbine are nine feet in diameter and keyed to the

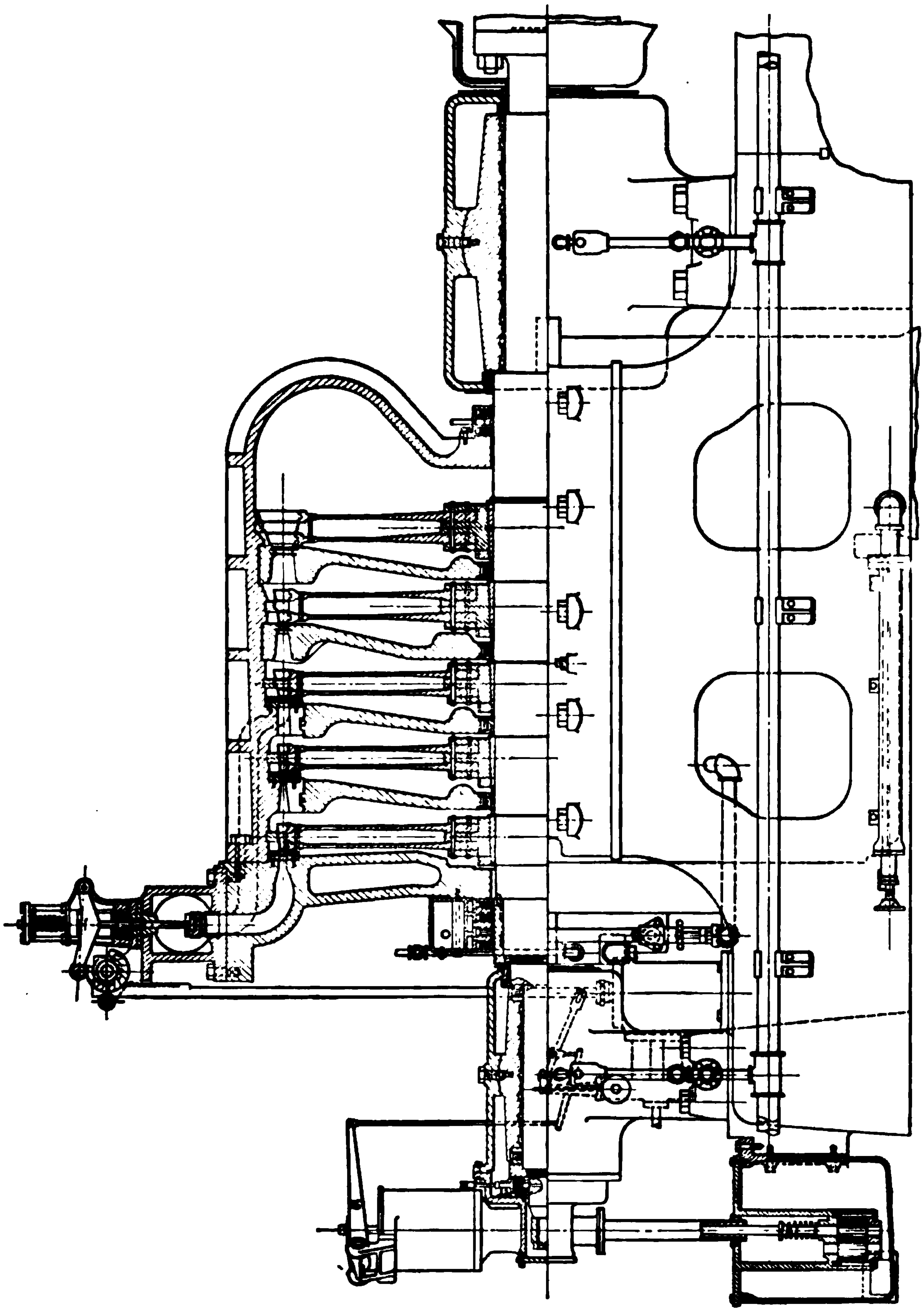


FIG. 179. Longitudinal Section through a 3500-Kw. Horizontal Curtis Steam Turbine.

vertical shaft S. On the periphery of each wheel are bolted two rows of blades or vanes, with a stationary or intermediate row attached to the casing between them. The buckets are made of rolled nickel bronze, hammered to shape and finish. The roots are dovetailed into the holders and the tips are tenoned and riveted into a shroud ring, thus insuring positive spacing and a rigid construction. Between each pair of wheels is a stationary steam-tight diaphragm P, which contains the nozzles through which the steam is expanded from the preceding stage. It will be noticed that the buckets and nozzles increase rapidly in size in succeeding stages as the pressure falls and the volume of steam increases.

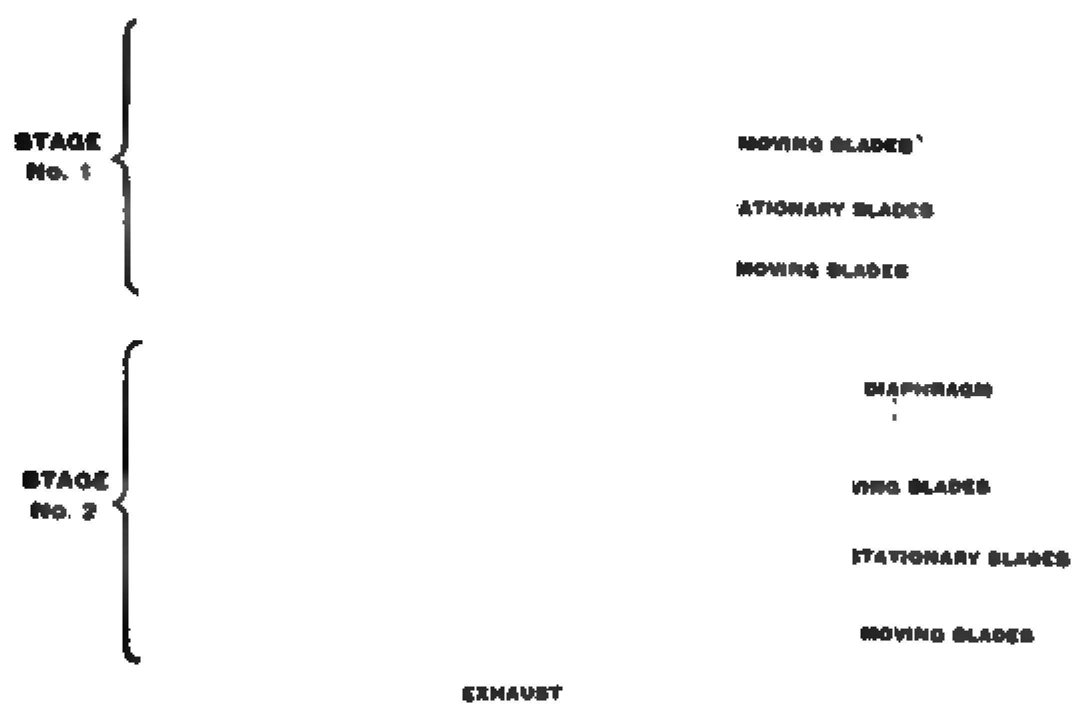


FIG. 180.

The parts are so proportioned that the steam gives up approximately $\frac{1}{n}$ of its energy in each stage, n representing the number of stages. The number of stages and the number of vanes in a stage are governed by the degree of expansion, the peripheral velocity which is desirable or practicable, and by various conditions of mechanical expediency. The number of admission valves vary in number and in location with the size of turbine. The automatic stage valve G connects the first stage directly to a set of auxiliary second-stage nozzles. Thus the overload capacity is increased by widening the steam belt and not by admitting high-pressure steam into an intermediate stage as was formerly the practice with Curtis turbines. This method of overload control results in higher efficiency than with the older system.

Curtis turbines appear to have a wider range of economical application than any other type, commercial sizes ranging from a small horizontal unit of 7 kilowatts rated output to vertical units of 20,000 kilowatts capacity on the continuous 24-hour basis. The smaller machines, 1000 kilowatts and under, are usually of the horizontal type, and the larger units, 3500 kilowatts and larger, are of the vertical type. Between 500 and 3500 kilowatts they are made both vertical and horizontal. All Curtis turbines are governed by "cutting-out nozzles"; that is, full initial pressure is maintained in all the nozzles that are open and

Fig. 181. Section through Curtis Governor.

the capacity of the machine is controlled by varying the number in operation. Units under 1500 kilowatts are ordinarily controlled by a *mechanical* valve gear and the larger units by an *indirect* or *relay* system. In the older types this relay system was electrically operated; in the modern machines the valves are hydraulically controlled.

Fig. 181 shows a section through a typical Curtis governor. Speed regulation is accomplished by the balance maintained between the centrifugal force of moving weights AA and the static force exerted by spring D. The governor is provided with an auxiliary spring F, for varying its speed when synchronizing, the tension in which is varied by

a small pilot motor controlled from the switchboard. The movement of the governor weights is transmitted through rod C to arm H and by means of the latter to the controlling mechanism of the valve gear.

Fig. 182 gives an assembly view of the mechanical valve gear as

FIG. 182. Assembly of Mechanical Valve Gears for 300-Kw. Curtis Steam Turbine.

applied to a 300-kilowatt unit. The valve stems extend upward through ordinary stuffing boxes and are attached to notched crossheads 8, 8. Each crosshead is actuated by a pair of reciprocating pawls or dogs, 6, 6, the lower one of which closes the valve and the upper one opens it. The several pairs of pawls are hung on a common shaft which receives a rocking motion from a crank driven by the turbine shaft. The cross-

heads have notches milled in the side in which the pawls engage to open or close the valve, the engagement being determined by shield plates 2, the positions of which are controlled by the governor through the medium of suitable levers. Shield plates 6 are set one a little ahead of the other to obtain successive opening or closing of the valves. The pawls are held in position when not in contact with the shield plates by springs *W*.

Fig. 183 gives a diagrammatic arrangement of the hydraulically controlled valve gear mechanism. The motion of governor *g* is transmitted through lever *i* to lever *a* of the pilot valve *j*. Pilot valve *j* controls the supply of oil (under pressure) in cylinder *k* the piston of which actuates rods *l*, *l*. The movement of rod *l* is transmitted through

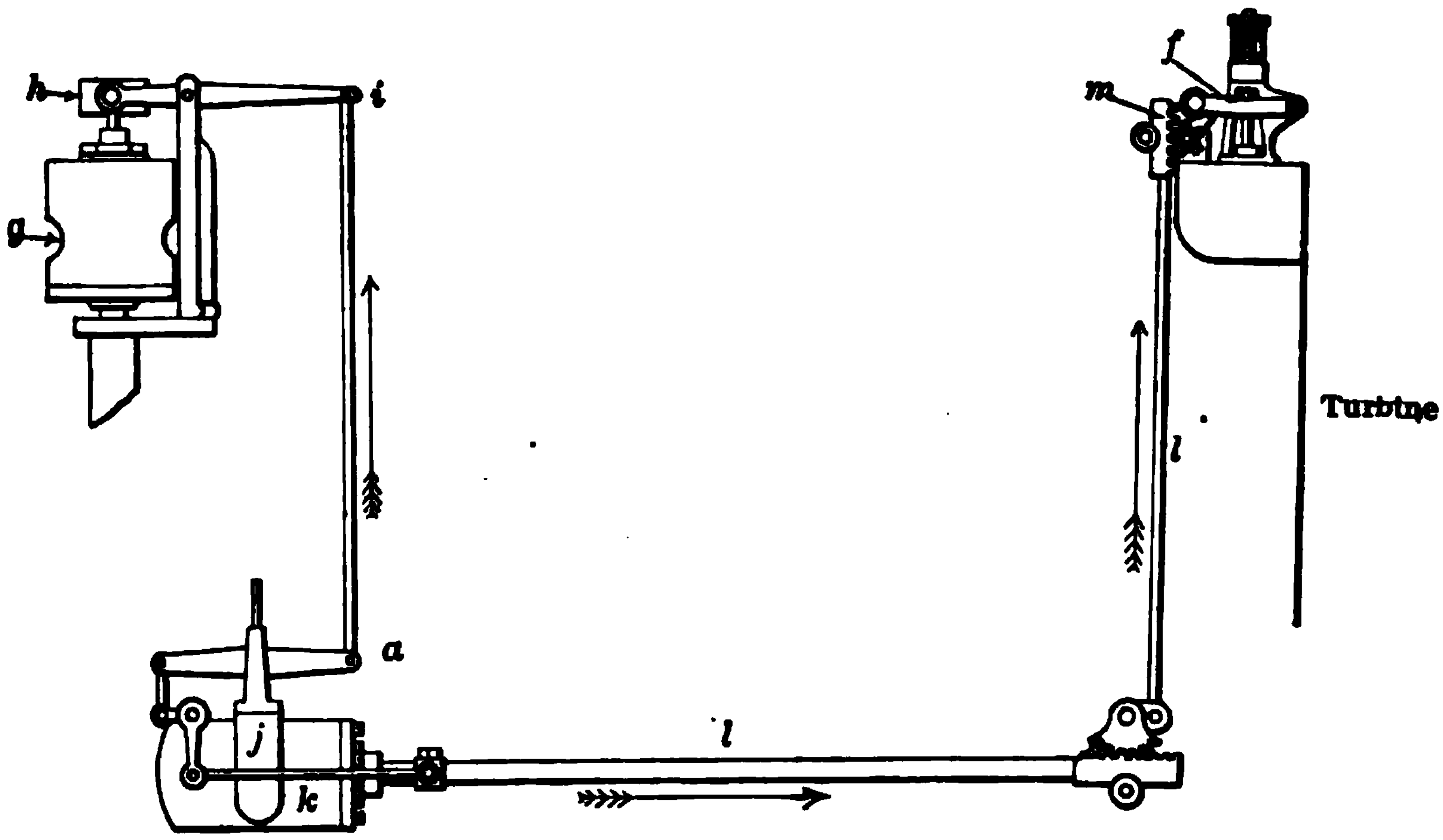


FIG. 183. Diagrammatic Arrangement of Hydraulically Operated Valve Gear, Curtis Turbine.

rack *m* to a small pinion. This pinion is mounted on the end of a shaft fitted with a number of cams, one a little ahead of the other, each cam controlling the opening and closing of a steam valve through the medium of rocker arm *f*. As the load on the turbine increases the governor slows down and causes the cam shaft to rotate in a reverse direction indicated by the arrow points in Fig. 183. This causes a proportionate number of valves to be lifted and held open, the number increasing as the load increases, until all are open. Should the load continue to increase, as in the case of overload, the secondary valve opens as previously described, connecting the first stage with a set of auxiliary second stage nozzles. Only the nozzles in the first stage are controlled by the governor. Should the turbine run above normal speed the emergency stop valve automatically closes the admission of steam to the nozzles. This device

consists of a steel ring placed around the shaft between the turbine and the generator. This ring is eccentrically mounted and the unbalanced centrifugal force is balanced by a helical spring. When the predetermined speed is reached the centrifugal force overcomes the spring tension and the ring moves in a still more eccentric position. In this position the ring strikes a bell crank lever which trips the throttle valve

Fig. 183a. Steam Belt Area in Five-Stage Curtis Turbine.

and permits it to close by its own weight and the unbalanced pressure on the valve stem.

In the Curtis turbine the area of the steam admission is limited to a small portion of the circumference in the first stages and does not

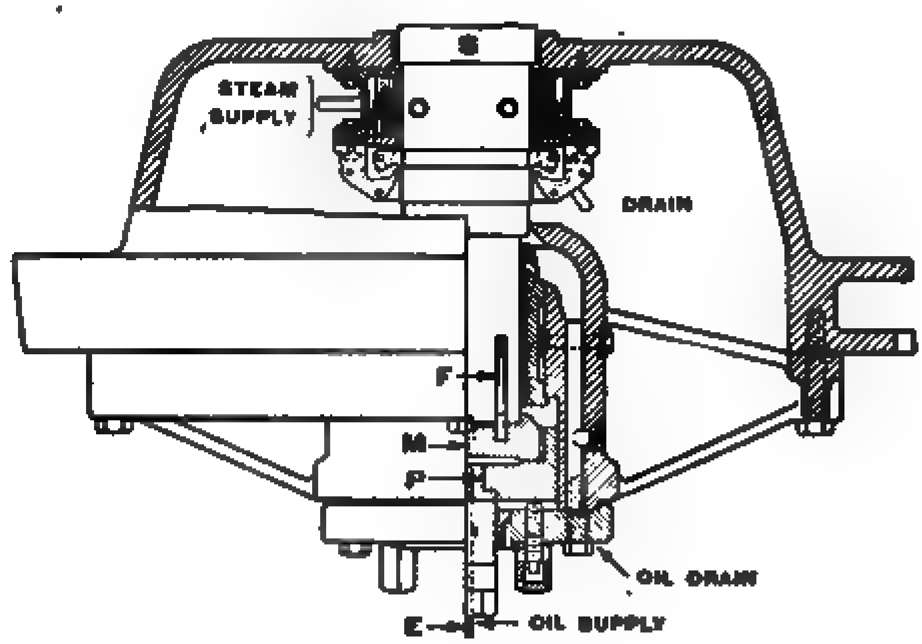


Fig. 183b. Step Bearing for Curtis Turbine.

extend around the entire circumference until the last stage is reached. See Fig. 183a.

The step bearing of a vertical machine is illustrated in Fig. 183b. The weight of the rotor is supported by oil under pressure forced between the bearing blocks *M* and *P*, thus permitting the shaft *S* to revolve on a film of oil. The smaller disk *M* is attached by dowels *F* to the main shaft. Carbon packing rings *O*, *O* are used above the bearing to prevent leakage, and adjustment is provided in the lower bearing block

by means of set screws. The oil pressure varies from 150 to 750 pounds per square inch according to the size of machine, the higher pressures being used in the larger machines.

Fig. 183c gives a diagrammatic outline of the oiling system. A tank, of sufficient capacity to contain all the oil and fitted with suitable straining devices and a cooling coil, is located at a level low enough to receive oil by gravity from all points lubricated. A pump draws oil from this tank and delivers it at a pressure about 25 per cent higher than that required to sustain the weight of the turbine in the step bearing. A spiral duct baffle connects the source of pressure to the step bearing and serves to regulate the oil supply to the lower end of the shaft. This source of pressure is also connected through a reducing valve to the upper oiling system of the machine, in which a pressure of about 60 lbs. to the square inch is maintained. This system, which includes a storage tank partly filled with compressed air, operates the hydraulic governor mechanism and supplies oil to the upper bearings. Delivery of oil to these bearings is regulated by adjustable baffles designed to offer resistance to the oil flow without forcing the oil to pass through any very small opening which might easily become clogged. A relief valve is provided to prevent the pressure in the upper part of the oiling system from rising above a desirable limit. Drain pipes from the upper bearings and from the hydraulic cylinder and relief valve all discharge into a common chamber, in which the streams are visible, so that the oil distribution can always be easily observed. At some point in the high-pressure system adjacent to the pump it is desirable to install a device to equalize the delivery of oil from the pump, as is done by the air chamber commonly used with pumps designed for low pressure. A small spring accumulator is furnished for this purpose, except in cases where weighted storage accumulators are used. In large stations where several machines are installed, a storage accumulator is desirable and can advantageously be so arranged that it will normally remain full, but will discharge if pressure fails, and in doing so will start auxiliary pumping apparatus.

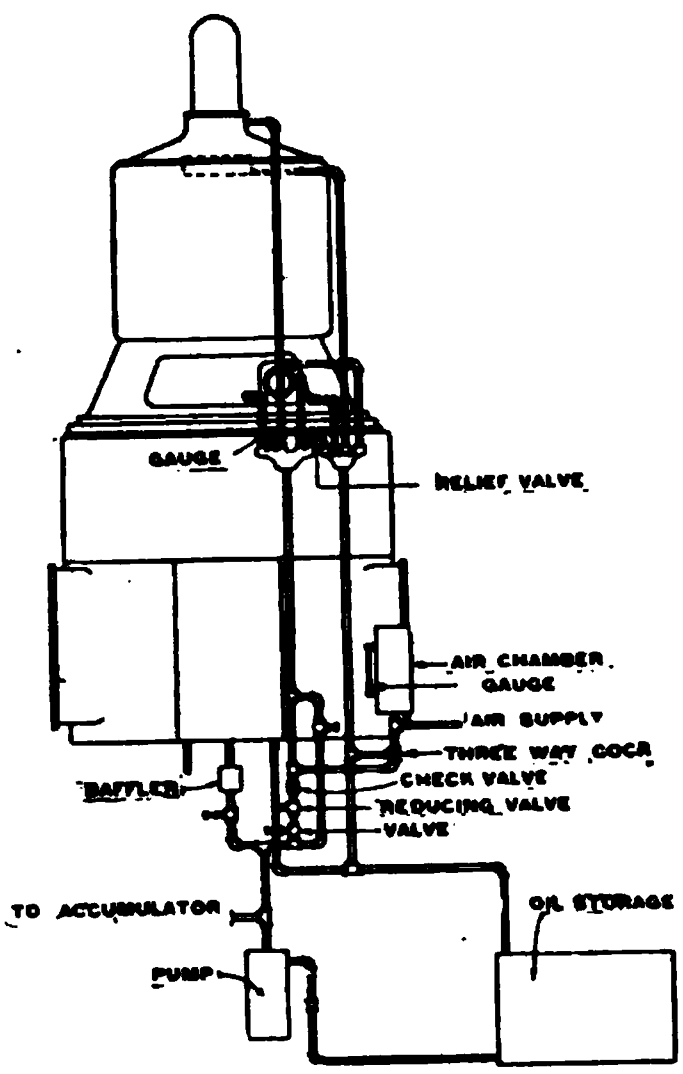


FIG. 183c. Arrangement of Oiling System for Curtis Turbine.

DIRECT CURRENT.

Kw.	R.p.m.	Kw.	R.p.m.
15	4,000	150	2,000
25	3,600	300	1,800
75	2,400	500	1,500

ALTERNATING CURRENT.

300	1,800	2,000	900
500	1,800	3,000 to 20,000	600–750
1,000	1,200		
1,500	900		

For the description of a typical steam turbine station equipped with Curtis turbines see Chapter XX.

General Description of Curtis Turbines: Power, March, 1909; Engr. U. S., Jan. 1, 1908, p. 115; Power & Engr., Feb. 25, 1908, p. 284, Feb. 25, 1908, March 3, 1908; Elec. Wld., June 17, 1905, p. 1136.

Guide Bearings, Oil Distribution and Carbon Packing: Power & Engr., April 14, 1908.

Mechanical Valve Gear: Power & Engr., March 10, 1908, p. 356.

Hydraulic Valve Gear: Power, March, 1909, p. 189.

190. Elementary Theory, Curtis Turbine. — Fig. 184 gives a diagrammatic arrangement of the blades and nozzles in the first stage of a two-stage Curtis turbine, each stage consisting of one set of nozzles and two moving and one stationary sets of blades. The object of employing a number of stages is to permit of a low peripheral velocity without reducing the efficiency. Since the velocity of steam varies as the square root of the kinetic energy, the theoretical stage velocity may be determined by dividing the maximum initial velocity by the square root of the number of stages, assuming that the entire velocity of the jet is abstracted in each stage. Thus if the turbine were constructed with four stages the theoretical stage velocity would be reduced from say 3600 feet per second to 1800 feet per second. In general, in order to reduce the stage velocity to V_s feet per second, the number of stages n may be determined from the equation

$$n = \frac{V^2}{V_s^2}, \tag{77}$$

in which V = maximum initial velocity.

Referring to Fig. 184: the steam is expanded in the first stage from pressure P_1 to P_2 and issues from the first set of nozzles with *absolute* velocity V_1, striking the first set of moving blades at an angle α with the line of motion of the wheel. The resultant v_1 of V_1 and the

peripheral velocity u, is the velocity of the steam *relative* to the vanes; and the angle β which the line v_1 makes with the line of motion of the wheel is the proper entrance angle of the blades for the first set. Neglecting friction the exit angle γ will be the same as the entrance angle β. The resultant of v_2 the exit velocity *relative* to the blade, and u, the peripheral velocity, is V_2, the *absolute* exit velocity.

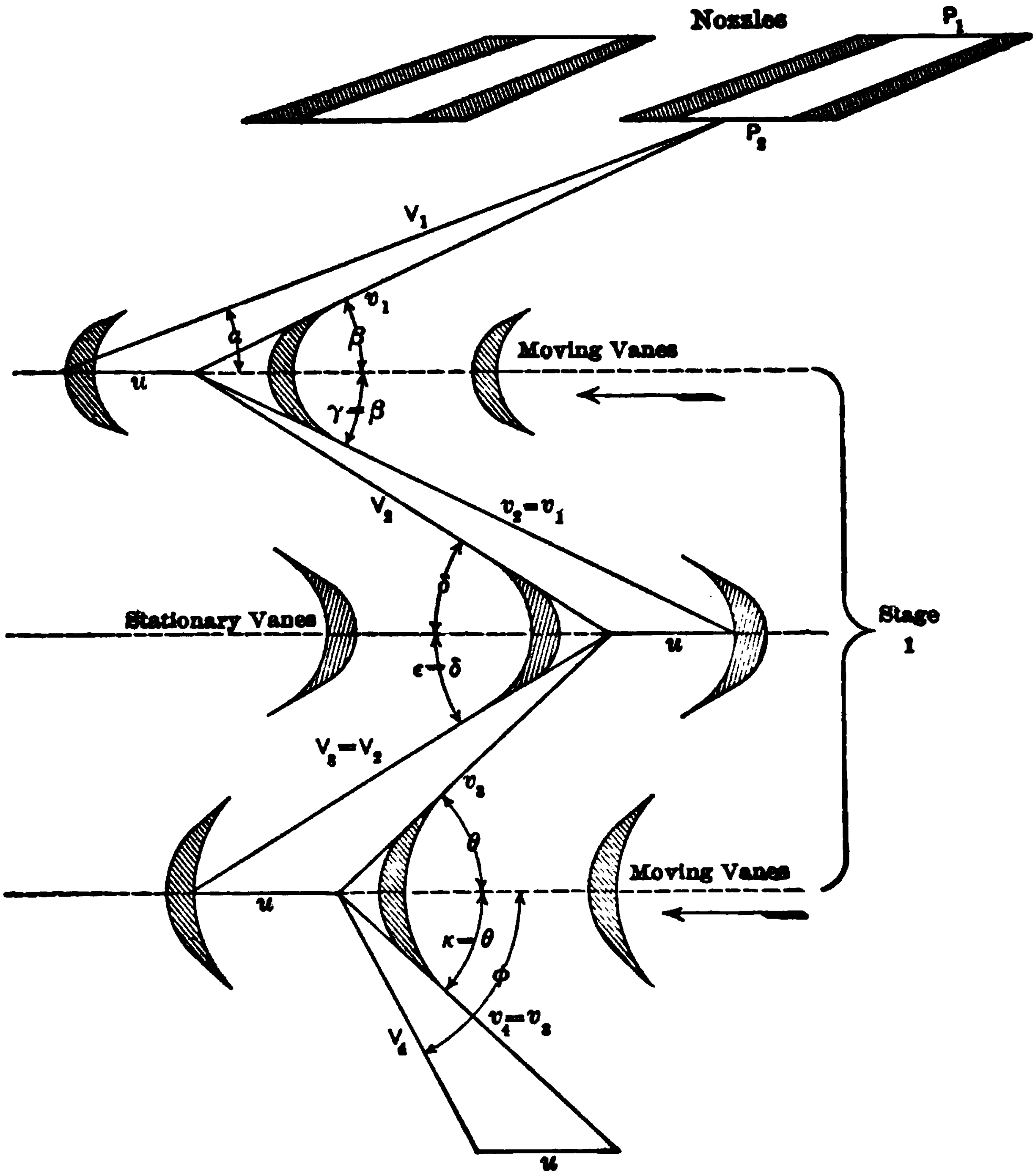


FIG. 184. Velocity Diagram, Curtis Turbine.

Since the second set of blades is fixed and serves as a means of changing the direction of flow, the absolute velocity entering them is V_2. The angle δ formed by V_2 and the center line of the stationary blades is the proper entrance angle. Neglecting friction the absolute exit velocity will be $V_3 = V_2$, and the exit angle will be $\varepsilon = \delta$. The steam flowing from the stationary blades strikes the second set of moving blades at an angle $\varepsilon = \delta$ with *absolute* velocity V_3. Combining V_3 with the

peripheral velocity u we get v_3, the velocity of the steam *relative* to the second set of moving blades. The angle θ, formed by v_3 and the line of motion of the wheel, is the proper entrance angle for the second set of moving blades. The resultant of v_4 ($= v_3$) and u is V_4, the absolute exit velocity for the first stage.*

In the second stage the steam is expanded from pressure P_2 to that in the condenser and acquires initial velocity V_a, leaving the last bucket with residual velocity V_n. The theoretical velocities and blade angles for this stage may be found as above.

Example: A two-stage Curtis turbine develops 800 horse power on a steam consumption of 12.5 pounds per horse-power hour, steam dry and saturated. Initial gauge pressure 135 pounds per square inch and back pressure 2 pounds per square inch absolute. Peripheral velocity 600 feet per second. The steam expands in the first stage from 150 pounds to 20 pounds absolute, and in the second stage from 20 to 2 pounds absolute. Angle of the nozzles with the plane of rotation, 20 degrees. Compare the performance of the actual turbine with its theoretical possibilities.

Actual turbine:

Steam consumed per hour = 800 × 12.5 = 10,000 pounds.

Steam consumed per second = 10,000 ÷ 3,600 = 2.78 pounds.

Horse power developed per pound of steam flowing per second,

$$800 \div 2.78 = 288.$$

Kinetic energy = 288 × 550 = 158,400 foot-pounds per second.

Thermal efficiency, equation (68),

$$E_t = \frac{2545}{12.5\,(1191.1 - 94.4)} = 18.7 \text{ per cent.}$$

Heat consumption, B.T.U. per horse power per minute,

$$\frac{12.5\,(1191.1 - 94.4)}{60} = 227.$$

$$\text{Efficiency ratio} = \frac{E_t}{E_r} = \frac{18.7}{25.3} = 0.739.$$

Ideal turbine:

Stage velocities.

The theoretical velocity in the first stage in expanding from a pressure of 150 pounds to 20 pounds absolute will be

$$\begin{aligned} V_1 &= 224\sqrt{H_1 - H_2} \\ &= 224\sqrt{1191.1 - 1042.9} \\ &= 2727 \text{ feet per second.} \end{aligned}$$

* In the actual turbine the velocities will be less than the theoretical on account of frictional resistances in the nozzles and blades, and the velocity diagram must be modified as indicated in Fig. 172b and described on p.

In the second stage the steam expands from 20 pounds to 2 pounds absolute.

$$V_a = 224\sqrt{H_3 - H_4}^{*}$$
$$= 224\sqrt{1042.9 - 914.8}$$
$$= 2530 \text{ feet per second.}$$

The kinetic energy per pound of steam in each division of the first stage will be:

In the first set of moving blades,

$$E = \frac{1}{64.4}(V_1^2 - V_2^2)$$
$$= \frac{1}{64.4}(2727^2 - 1620^2) = 74{,}722 \text{ foot-pounds per second.}$$

The value of V_2 is conveniently obtained from the velocity diagram.

In the second set of moving blades,

$$E = \frac{1}{64.4}(V_2^2 - V_3^2)$$
$$= \frac{1}{64.4}(1620^2 - 925^2) = 27{,}448.$$

Total energy in first stage = 102,170.

In a similar manner the total energy in the second stage will be found to be 93,700.

Total for entire turbine = 195,870 foot-pounds per second.

Theoretical horse power per pound of steam:

$$\text{H.P.} = \frac{195{,}870}{550} = 356.$$

Theoretical steam consumption per horse-power hour,

$$\frac{3600}{356} = 10.1 \text{ pounds.}$$

Heat consumption, B.T.U. per horse power per minute,

$$= \frac{10.1\,(1191.1 - 94.4)}{60} = 184.$$

Thermal efficiency ratio,

$$E_r = \frac{1191.1 - 914.8}{1191.1 - 94.4} = 0.253.$$

* In the actual turbine the heat contents H_2, H_3 and H_4 will be greater than that of the ideal mechanism on account of frictional losses. See equation (76d).

SUMMARY.

	Actual Turbine.	Perfect Turbine.
Horse power developed per pound of steam	288	356
Steam consumption, pounds per H.P. hour	12.5	10.1
B.T.U. consumed per H.P. per minute	268	184
Thermal efficiency, per cent	18.7	25.3
Efficiency ratio, per cent	73.9	....

191. The Hamilton-Holzworth Steam Turbine. — Figs. 185 to 188 give a general view and some of the details of the Hamilton-Holzworth turbine, which belongs to the compound multi-stage "velocity" type. The steam flows through the annular space between rotor and stator

Fig. 185. Principles of Hamilton-Holzworth Steam Turbine.

as in the Parsons, but differs from the latter in that expansion takes place only in the stationary vanes. The rotor consists of a number of steel disks of varying diameters riveted to both sides of steel hubs and fitted at the periphery with drop-forged vanes as shown in Fig. 186. A tough steel ring is shrunk on the outside periphery of the vanes as indicated. The number of wheels and vanes is considerably less than in the Parsons type. The stationary vanes are fitted in steel disks as shown in Fig. 185, and the latter are located in grooves in the turbine casing. The vanes have a varying radial height increasing in the direction in which the steam flows. Sizes under 750 kilowatts

have but one turbine casing, but larger sizes are divided into two, a high and a low-pressure turbine. The operation is as follows: Steam enters the high-pressure casing as indicated by arrows in Fig. 185, and passes through the first set of stationary vanes, extending around the whole periphery, which direct the steam at the proper angle against the wheel blades. In passing through the stationary vanes the steam is expanded down to the pressure in the first stage which is the same on both sides of the rotating disk. After giving up part of its energy, the steam expands again through the second set of stationary vanes to the pressure in the second stage, giving up energy to the second set of moving vanes. This process is repeated until the last stage is reached, from which the steam is discharged to the condenser in the simple turbine, or to the low-pressure steam chest in the compound turbine. In the low-pressure casing the steam is distributed in the same manner as in the high-pressure turbine. The diagram in the lower part of Fig. 185 shows the variation in steam pressure and velocity. The low-pressure front head is provided with an auxiliary nozzle which may be supplied with live steam in case of overload. The builders claim that since the pressures on both sides of the wheel are the same, no provision is necessary for axial balancing as in the Parsons standard turbine.

SECTION OF WHEEL.

FIG. 186. Details of Vanes, Hamilton-Holsworth Steam Turbine.

Fig. 187 shows a sectional view of the bearing and stuffing box for the shaft at the point where it passes through the end of the turbine casing. The shaft is turned to a smaller diameter at its end and runs in a bushing *G* having a flange bearing against the inner side of the pillow block. At *A* is a cylindrical piece attached to and rotating with the shaft. This piece projects into an annular groove in the piece *B*, but it does not completely fill the groove and a circuitous passage is formed through which the steam must pass before reaching the stuffing box *C*. The object of the passage is to provide condensing

surface so the steam itself will not reach the packing. The joint at the stuffing box is thus practically water-sealed.

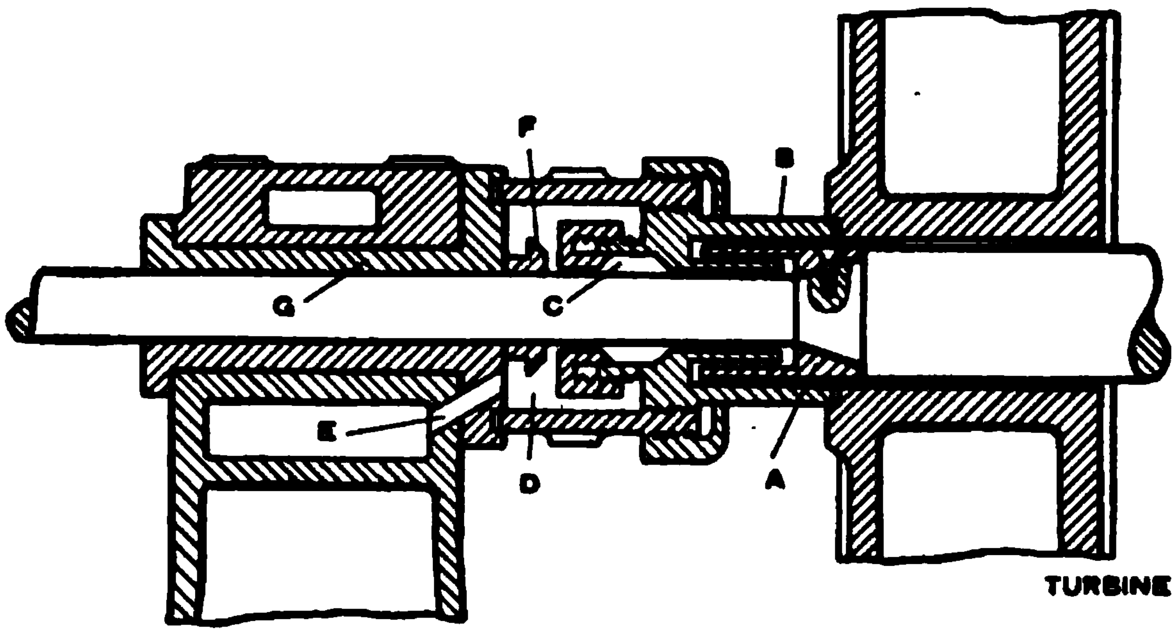


FIG. 187. Details of Bearing, Hamilton-Holzworth Turbine.

To prevent the oil from working into the turbine a bushing F is attached to the shaft which throws off the oil into the space D by centrifugal force, where it drips down through a channel into a compartment in the pillow block. Any water escaping through the stuffing box is also collected in the same compartment. The bearing is oiled by a forced-oil system, the oil being supplied to the bottom of the bushing.

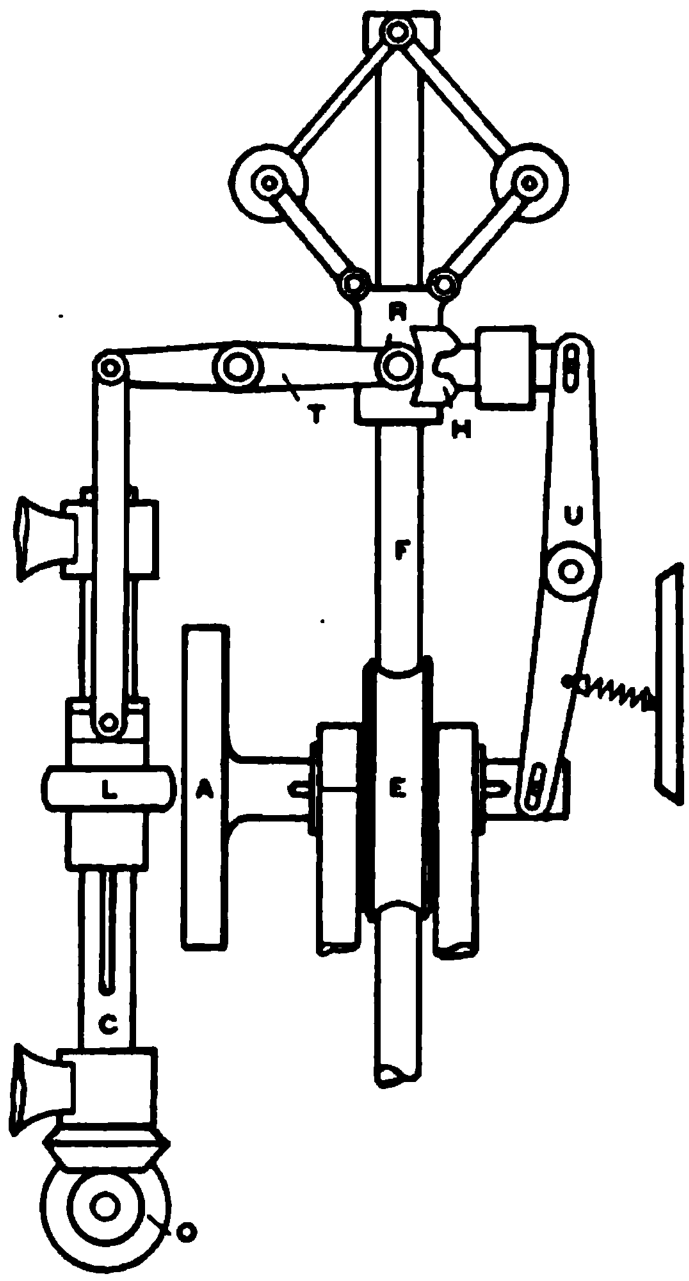


FIG. 188. Governor Mechanism, Hamilton-Holzworth Turbine.

Fig. 188 gives a diagrammatic view of the governor mechanism. A is the friction disk, L the roller, C the splined shaft with which the roller turns but upon which it is free to slide, O bevel gears connecting shaft C with the throttle valve, and E a worm wheel driving disk A by means of a worm shaft F. At normal speed the governor sleeve is in mid position and roller L is at the center of the disk. If the turbine speeds up, however, the governor sleeve will rise, carrying with it the right-hand arm of lever T, which in turn will push the roller L a corresponding distance downward. At the same time the cam H will be thrown to the right by contact with the roller R and by means of lever U will move the disk A and its shaft to the left, bringing it in

contact with roller L, thus imparting a rotary motion to the shaft C and closing the throttle valve until the turbine assumes normal speed, when the several parts assume the first position. If the speed is reduced below normal the operation is just the same except that the various motions are reversed.

At an increase in speed of 2½ per cent above normal steam is cut off entirely.

Hamilton-Holzworth Turbine: Am. Elecn., Oct., 1904, p. 549; Eng. Rec., Oct. 1, 1904, p. 405; Power, Dec., 1907, p. 878, Nov., 1904, p. 659; Machinery, Nov., 1904, p. 134; Engr. U.S., Oct. 1, 1904, p. 690.

192. Westinghouse-Parsons Steam Turbine.—Fig. 189 shows a section through a Westinghouse-Parsons multi-stage reaction turbine. In this type no nozzles are employed and expansion of the steam is effected by a series of stationary and movable blades. The rotor is a steel barrel or drum divided into three sections of varying diameter, upon the periphery of which bronze blades are radially inserted in dovetailed grooves. The adoption of three sections of varying diameter has no bearing on the design of this machine but is merely for mechanical convenience. The blades increase in length and cross section from the high-pressure to the low-pressure end of each section. The stator is of cast iron and its inner surface is studded with rows of blades projecting radially inward and conforming in size with the adjoining blades of the rotor. The relative positions of the blades in the rotor and stator are shown in Fig. 190. The operation of the turbine is as follows: Steam enters at S, Fig. 189, through poppet valve V, which is actuated by the governor shown in detail in Fig. 191, and flows through the annular space between rotor and stator to the exhaust opening at B. The entire expansion is carried out within this annular compartment and resembles in effect a simple divergent nozzle with the exception that the dynamic relationship of jet and vane is such as to secure a comparatively low velocity from inlet to exhaust. The velocity varies from 150 feet per second at the high-pressure end to about 600 feet per second as a maximum at the low-pressure end. The action of the steam on the blades is illustrated in Fig. 190. The steam strikes the first set of stationary blades as at P with initial velocity of about 150 feet per second and is deflected against the moving blades immediately adjoining. In passing from P to P_1 the steam is partly expanded and gives up a portion of its energy to the moving blades. The steam is deflected from P_1 to P_{11} and thus has a reactive effect on the moving blades in addition to the impulse imparted at P_1. The total torque produced at the shaft in element A is therefore due to impulse from

FIG. 189. Section through Westinghouse-Parsons Standard Steam Turbine.

1 and reaction from 2. This process is repeated in each element of the turbine, the steam expanding as it flows from element to element in its passage to the condenser. The angular velocity of the rotor

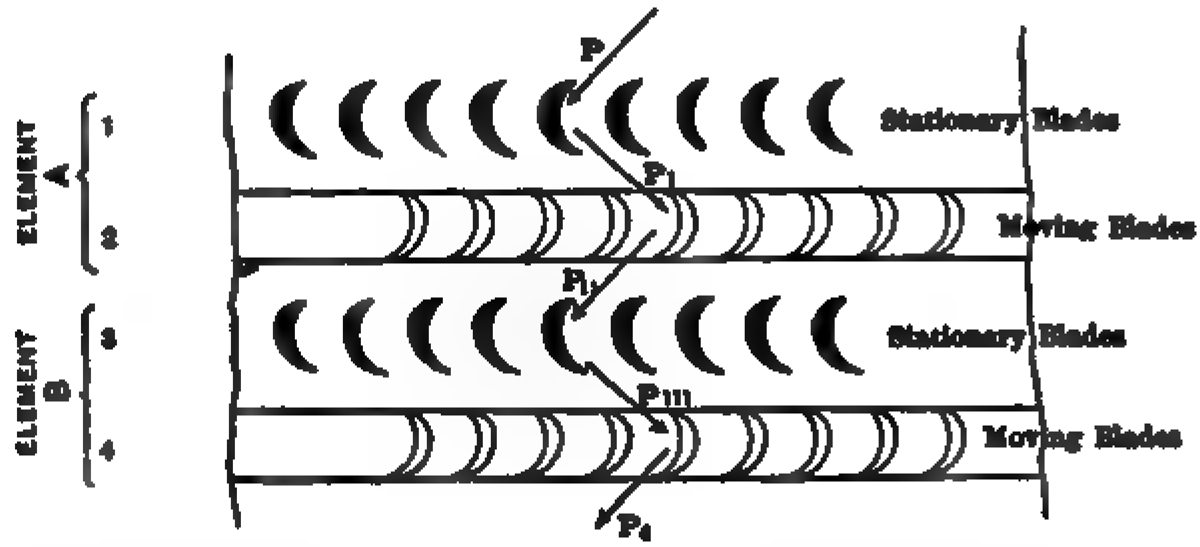


FIG. 190. Flow of Steam in Parsons Turbine.

varies from 3600 r.p.m. in a 400-kilowatt unit to 750 r.p.m. in the 7500-kilowatt size. Opposed to the three sets of blades the spindle also carries three rotating balance pistons *P*, *P*, Fig. 189, each of such diameter as

FIG. 191. Governor Mechanism, Westinghouse-Parsons Turbine.

to exactly balance, through passage *E*, the axial thrust of the steam against its corresponding drum of blades.

Steam enters the turbine intermittently as shown in Fig. 192, which represents indicator cards from a 1250-kilowatt turbine at various loads.

At light load the valve opens for a very short period and remains closed during the greater part of the interval. As the load increases, the period lengthens until finally, at about full load, the valve does not reach its seat at all, and continuous pressure is obtained in the high-pressure end of the turbine.

The intermittent admission of steam is produced and controlled as follows: Lever T, Fig. 191, is given a reciprocating motion by an eccentric actuated by a worm and worm wheel on the main shaft. This motion is transmitted through lever H (with fixed fulcrum B) to lever A (with floating fulcrum D) and finally to pilot valve G. This reciprocating pilot valve admits puffs of steam from pipe O to the under side of piston M, the rod R of which is attached to the admission valve V in Fig. 189. A spiral spring holds piston M in its lowest position until

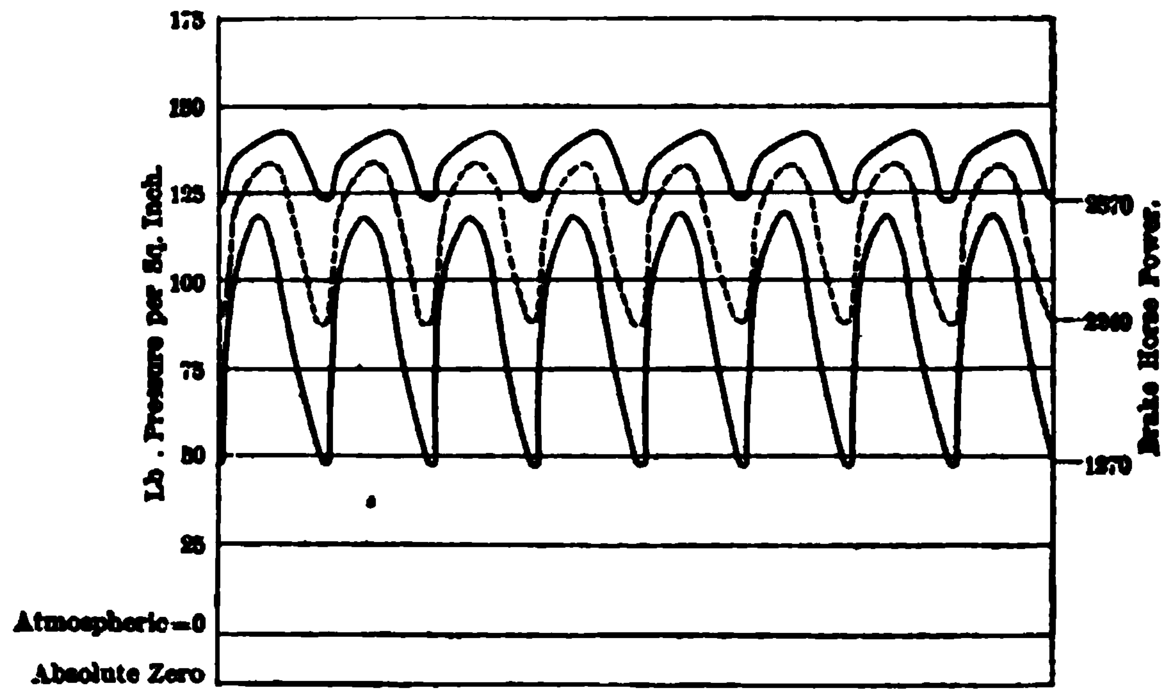


FIG. 192. Indicator Cards Showing Initial Pressure in a Westinghouse-Parsons Steam Turbine.

steam admitted by the pilot overcomes the spring tension and lifts the main valve from its seat, thereby permitting steam to enter the turbine. The fulcrum D of lever A is raised and lowered by the governor and therefore the pilot valve is controlled both by the motion of the eccentric and the motion of the governor. The eccentric keeps the pilot valve, and hence the main throttle, in constant oscillation, while the movement of the governor changes the limits of this motion.

If an overload is sufficiently great to cause the governor balls to drop to their lowest position, the auxiliary or secondary valve V_s, Fig. 189, begins to open and admits high-pressure steam to the later stage where the working steam areas are greater, thus increasing in proportion the total power of the turbine. The operation of this valve is the same as the main admission valve and is controlled by the governor. Fig. 193 shows the details of this mechanism. The speed varies about 2 per cent from no load to full load.

In the smaller size machines running above 1200 r.p.m., flexible bearings are employed to absorb the vibration incident to the *critical* velocity. They consist of a nest of loosely fitting concentric bronze sleeves with sufficient clearance between them to insure the formation of a film of oil. In the larger machines running below 1200 r.p.m. a split self-aligning bearing is used instead of the flexible bearing. The ends of the casing are fitted with water-sealed glands of special design to prevent the escape of steam or inflow of air at the point of entry of the shaft. The water used for sealing them is small in quantity and may be returned to the feed-water system.

FIG. 193. By-Pass Valve, Westinghouse-Parsons Turbine.

Double-flow Type. — In reaction turbines of the single-flow type, as illustrated in Fig. 189, the high-pressure portion dealing with the high-pressure incoming steam is the least efficient. This is due to the fact that the blade lengths are approximately proportional to the specific volume of the steam, and consequently the initial expansion in the turbine requires blade passages of very small dimensions. This results in greater leakage past the tips of the blades than in the low-pressure elements where the blades are long. Again, in the single-flow type the high-pressure balance piston occupies fully one-half of the total balance piston length of the shaft, while the low-pressure piston is $2\frac{1}{2}$ times the high-pressure diameter, so that balance pistons occupy a large portion of the total bulk of the machine. By making the high-pressure element of the impulse type and by arranging the low-pressure reaction elements on either side as illustrated in Fig. 193b the efficiency may be increased and the bulk of the turbine may be greatly decreased.

There are two rows of moving buckets upon the impulse wheel with an intermediate set of reversing blades, the operation being practically the same as in the first stage of a Curtis turbine. The drop in pressure in the nozzles is such that approximately 20 per cent of the total energy developed is absorbed by this impulse element. After leaving the impulse element the steam divides, one portion passing directly to the low-pressure blading at the left, while the rest passes through the hollow shell of the rotor to the similar pressure blades upon the right. As these sections are equal and symmetrical they counterbalance each other, and the balance or " dummy " pistons may be dispensed with. The advantages of the double-flow type over a single-flow unit of equal capacity are: (1) reduction of nearly 50 per cent in the shaft span

Fig 193a. Method of Fastening Blades in Westinghouse-Parsons Turbines.

between bearings; (2) the diameters of the casing and rotating part are more uniform, thus tending to greater rigidity; (3) a reduction of about 70 per cent in the bulk of the main parts of the machine, and (4) internal stresses due to high-pressure and high-temperature steam are avoided by isolating the incoming steam, without separate nozzle chambers. Westinghouse-Parsons turbines are made in a number of sizes, varying from 400 kw. to 15,000 kw. In Europe, however, Parsons turbines are made as small as 20 kw. and as large as 25,000 H.P. In sizes up to 3500 kw. the single-flow turbine has established itself as the most suitable prime mover, but for larger sizes the double-flow is given preference The double-flow turbine is admirably adapted to low-pressure work. Fig. 194b shows a section through a Westinghouse-Parsons double-flow low-pressure turbine. For results of tests of Parsons and Westinghouse-Parsons turbines see Table 48.

Double-flow Turbine: Power & Engr., March 16, 1909, Aug., 1908, p. 471; Eng. Rec., May 30, 1908, p. 693; Elec. Review, June 26, 1908, p. 1089.

10,000-kw. Westinghouse-Parsons Double-flow Turbo Generator for the Metropolitan Street Railway Company, Kansas City, Kansas: Power & Engr., May, 17, 1910, p. 890.

Nozzle Block

Fig. 193b. Westinghouse-Parsons Double-flow Turbine with High-pressure Impulse Stage.

192a. Allis-Chalmers Steam Turbine. — Fig. 193c shows a section through an Allis-Chalmers standard steam turbine, which is of the Parsons type but differs from the original Parsons machine and the Westinghouse-Parsons construction principally in manufacturing details. In the older Parsons type, three balance pistons are placed at the high-pressure end. In the Allis-Chalmers design, the larger piston is placed at the low-pressure end of the rotor, behind the last row of blades, the other two remaining at the high-pressure end. This construction permits of a smaller balance piston and allows a smaller working clearance in the high-pressure and intermediate cylinders. In the Allis-Chalmers turbine the roots of the blades are dovetailed and fitted into a foundation ring, and the tips are encased in a channel-shaped shroud ring, thereby insuring a rigid and positively spaced construction. The governor is of the Parsons type, except that the main valve and pilot valve are actuated by hydraulic instead of steam pressure. The bearings are of the self-adjusting ball and socket pattern and are kept "floating in oil" by a small pump geared to the turbine shaft. The oil is passed through a tubular cooler with water circulation after it leaves the bearings and is used over and over again.

Fig. 193c. Section through Allis-Chalmers Steam Turbine.

193. Elementary Theory, Parsons Turbine. — Fig. 194 gives a diagrammatic arrangement of fixed and stationary blades in the first stages of a multi-stage ideal reaction turbine. The steam enters the stationary blades at practically zero velocity and is there partially expanded and impinges against the movable blades at velocity V_1, part of the energy of the steam being thus absorbed. In passing through the movable blades the steam is still further expanded and leaves at an absolute velocity V_2, exerting an additional pressure on the blades from the reaction. The steam enters the second set of stationary blades with velocity V_2 and is still further expanded to velocity V_3, and so on.

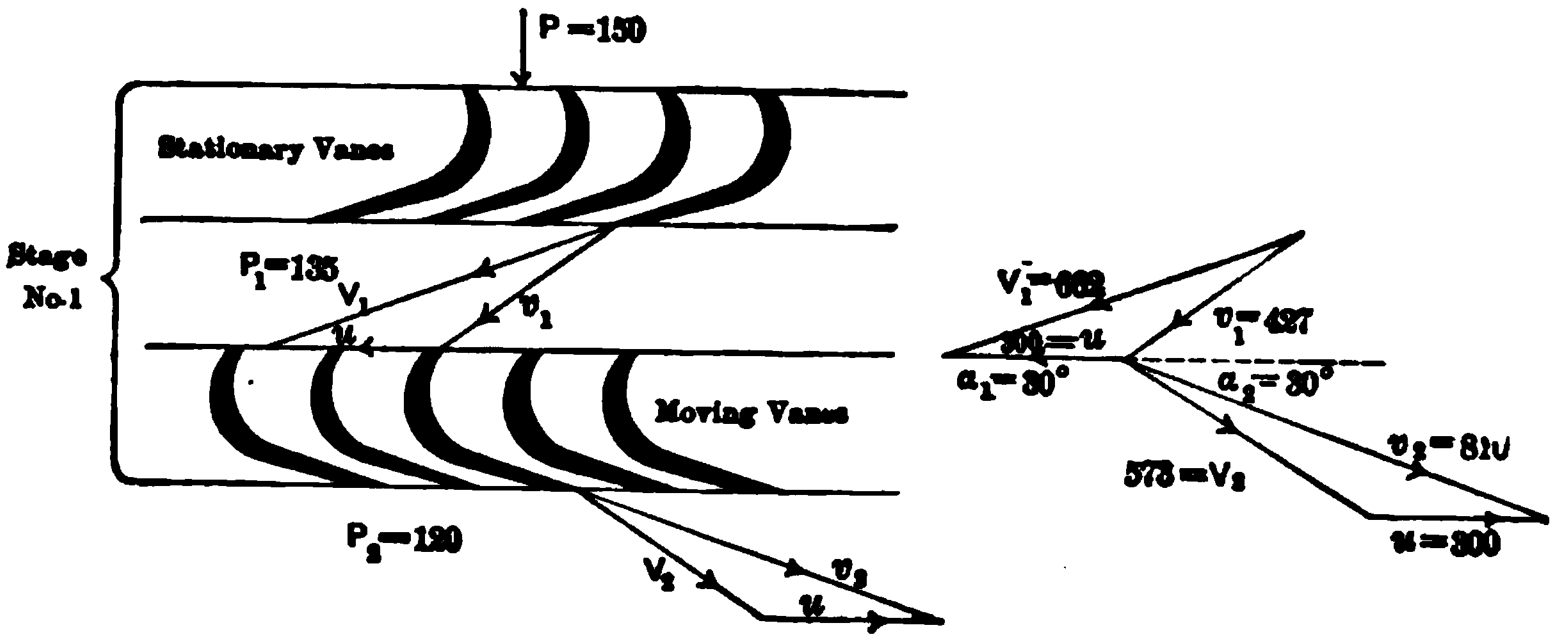


FIG. 194. Velocity Diagram. Westinghouse-Parsons Turbine.

The energy imparted to the steam in the first set of stationary blades is

$$E_1 = \frac{W}{2g} \cdot V_1^2. \tag{78}$$

V_1 = absolute velocity of the steam leaving the blades.

The energy imparted to the steam in the first set of moving blades is

$$E_2 = \frac{W}{2g}(v_2^2 - v_1^2). \tag{79}$$

v_1 = relative velocity of the steam entering the moving blades.

v_2 = relative velocity of the steam leaving the moving blades.

The total energy acquired by the steam in the first stage is

$$E_1 + E_2.$$

The energy converted into work in this stage is

$$E = E_1 + E_2 - \frac{WV_2^2}{2g} \tag{80}$$

$$= (V_1^2 + v_2^2 - v_1^2 - V_2^2)\frac{W}{2g}. \tag{81}$$

V_2 = absolute velocity of the steam leaving the moving blades.

Each stage may be analyzed in a similar manner.

Example: A Westinghouse-Parsons turbine develops 1000 horse power on a steam consumption of 12 pounds of steam per horse-power hour. Initial steam pressure 150 pounds per square inch absolute; back pressure 1 pound per square inch absolute; drop in pressure in each set of fixed and moving blades 15 pounds per square inch; peripheral velocity 300 feet per second; $\alpha_1 = \alpha_2 = 30$ degrees. Compare the performance of the actual and ideal turbine.

Actual turbine:

Steam consumed per hour,

$$1000 \times 12 = 12{,}000 \text{ pounds.}$$

Steam consumed per second,

$$12{,}000 \div 3{,}600 = 3.33 \text{ pounds.}$$

Horse power developed per pound of steam flowing per second,

$$1000 \div 3.33 = 300.$$

Kinetic energy per pound of steam,

$$300 \times 550 = 165{,}000 \text{ foot-pounds per second.}$$

Thermal efficiency,

$$E_t = \frac{2545}{12\,(1191.2 - 70)} = 18.9 \text{ per cent.}$$

Heat consumption, B.T.U. per horse-power hour per minute,

$$\frac{12\,(1191.2 - 70)}{60} = 224.$$

Efficiency ratio,

$$\frac{E_t}{E_r} = \frac{18.9}{28} = 67.5 \text{ per cent.}$$

Ideal turbine:

The velocity imparted to the steam in the first set of stationary blades due to the drop from 150 to 135 pounds per square inch is

$$\begin{aligned} V_1 &= 224\sqrt{H_1 - H_2} \\ &= 224\sqrt{1191.2 - 1182.4} \\ &= 662 \text{ feet per second.} \end{aligned}$$

Lay off the value of V_1 in direction and amount and combine with u, the peripheral velocity, Fig. 194. The resultant is v_1, the velocity of the steam relative to the blades. The angle between v_1 and the line of motion of the wheel will be the angle with the blade at entrance.

From the velocity diagram,

$$v_1 = 429.$$

E_2, the energy given up by one pound of steam in expanding from 135 to 120 pounds, is

$$
\begin{aligned}
E_2 &= 778\ (H_2 - H_3) \\
&= 778\ (1182.4 - 1172.8) \\
&= 7468 \text{ foot-pounds per second.}
\end{aligned}
$$

Substitute $v_1 = 429$ and $E_2 = 7468$ in equation (79),

$$7468 = \frac{1}{64.4}(v_2^2 - 429^2),$$

$$v_2 = 816 \text{ feet per second.}$$

The resultant of v_2 and u is V_2, the absolute velocity of the steam leaving the moving blades of the first stage. From the diagram,

$$V_2 = 573 \text{ feet per second.}$$

The energy converted into work in the first stage is determined by substituting the proper values in equation (81), thus:

$$
\begin{aligned}
E &= (\overline{662}^2 + \overline{816}^2 - \overline{429}^2 - \overline{573}^2)\frac{1}{64.4} \\
&= 9200 \text{ foot-pounds per second.}
\end{aligned}
$$

The various stages may be analyzed in a similar manner.

The theoretical output of the entire turbine per pound of steam will be that corresponding to adiabatic expansion from a pressure of 150 to 1 pound absolute.

$$
\begin{aligned}
E &= 778\ (H_1 - H_n) \\
&= 778\ (1191.2 - 877) \\
&= 244{,}447 \text{ foot-pounds per second.}
\end{aligned}
$$

Horse power per pound of steam,

$$\text{H.P.} = \frac{244{,}447}{550} = 445.$$

Steam consumption per horse-power hour,

$$\frac{3600}{445} = 8.1 \text{ pounds.}$$

Thermal efficiency,

$$
\begin{aligned}
E_r &= \frac{1191.2 - 877}{1191.2 - 70} \\
&= 28 \text{ per cent.}
\end{aligned}
$$

194. Low-pressure and Mixed-pressure Turbines. — A promising field for the steam turbine is in its application as a secondary or low-pressure unit in connection with non-condensing or condensing engines, or, combined with a regenerator, in connection with engines using steam intermittently. Numerous examples may be cited showing great gains in both capacity and economy in existing power plants involving the

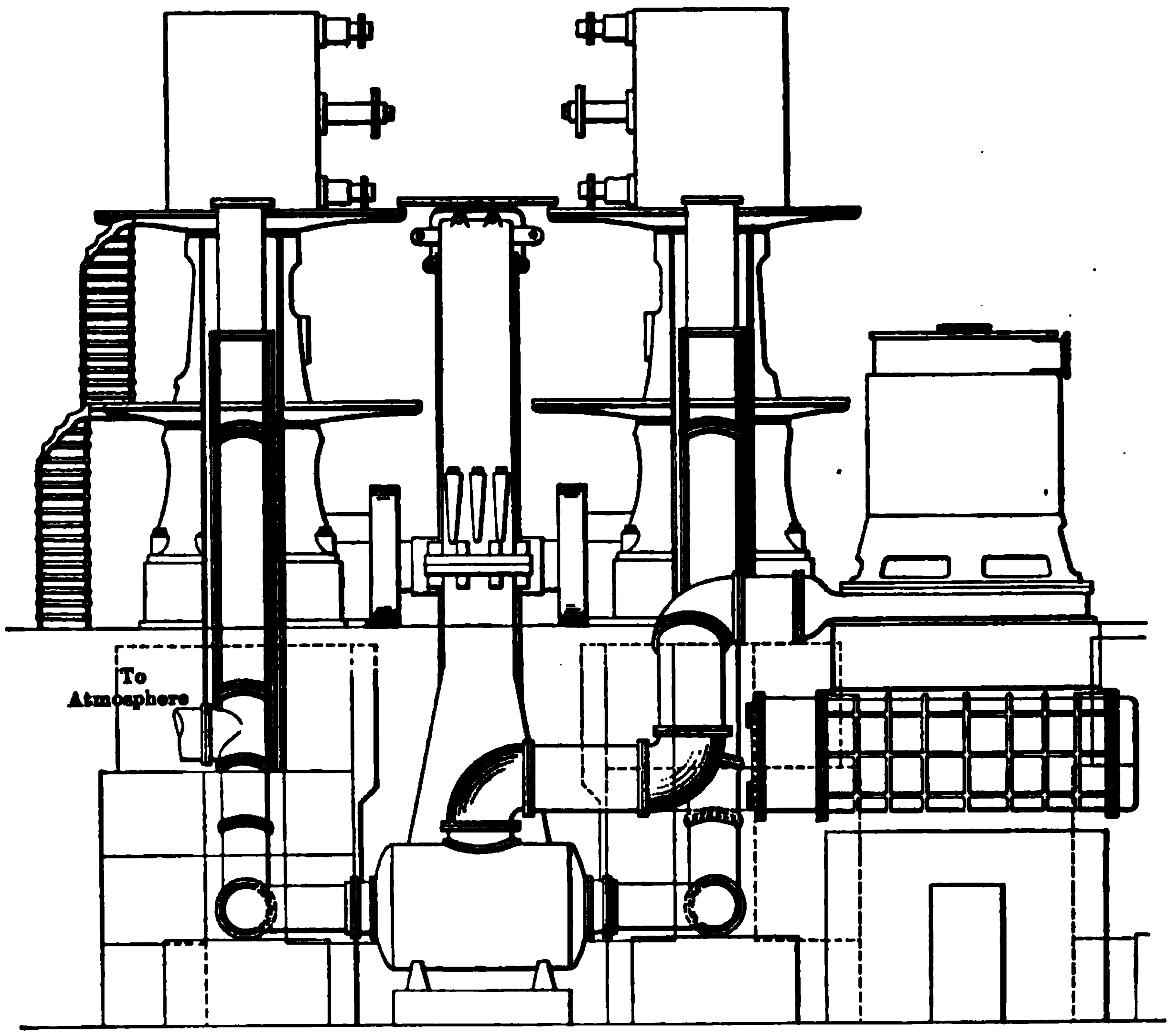


FIG. 194a. Low-pressure Turbine Installation at the 59th Street Station of the Interborough Rapid Transit Company, New York.

abandonment of but a negligible part of the equipment and accomplishing this result with a minimum additional investment. The most notable installation (June, 1910) of low-pressure turbines to condensing reciprocating engines is at the 59th Street Station of the Interborough Rapid Transit Co., New York. Three of the nine 7500-kw. Manhattan-type compound Corliss engines have been equipped with Curtis three-stage, low-pressure turbo-generators of equal capacity, and provision is made for the installation of six additional units. The low-pressure turbine is installed between the exhaust of the low-pressure cylinders and the condenser as shown in Fig. 194a. Running with the

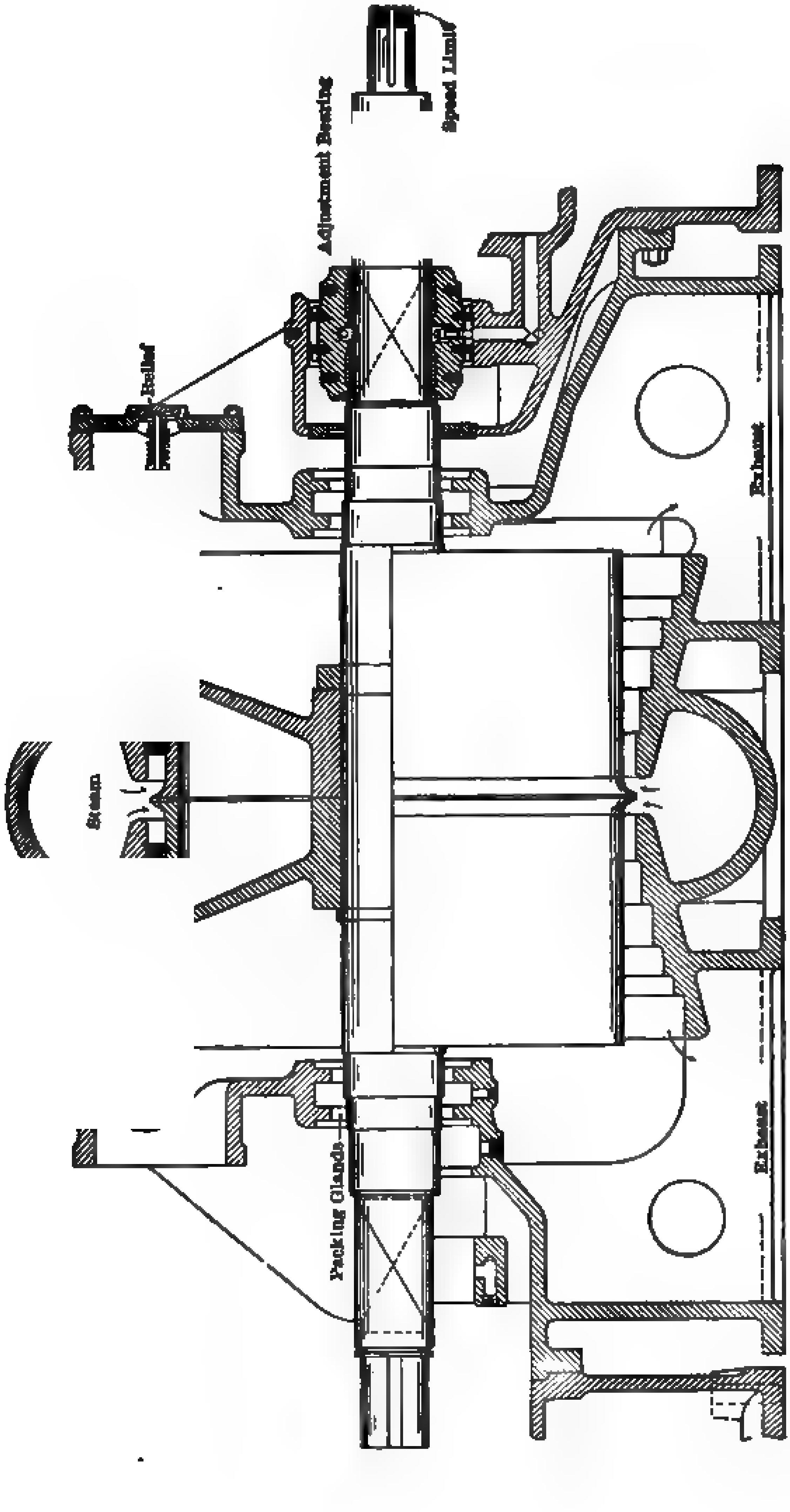


Fig. 194b. Westinghouse Double-flow Low-pressure Turbine.

engine the low-pressure turbine generator carries a variable load without governor regulation. The turbine generator takes care of the speed by automatically taking such a load as will keep the frequency in unison

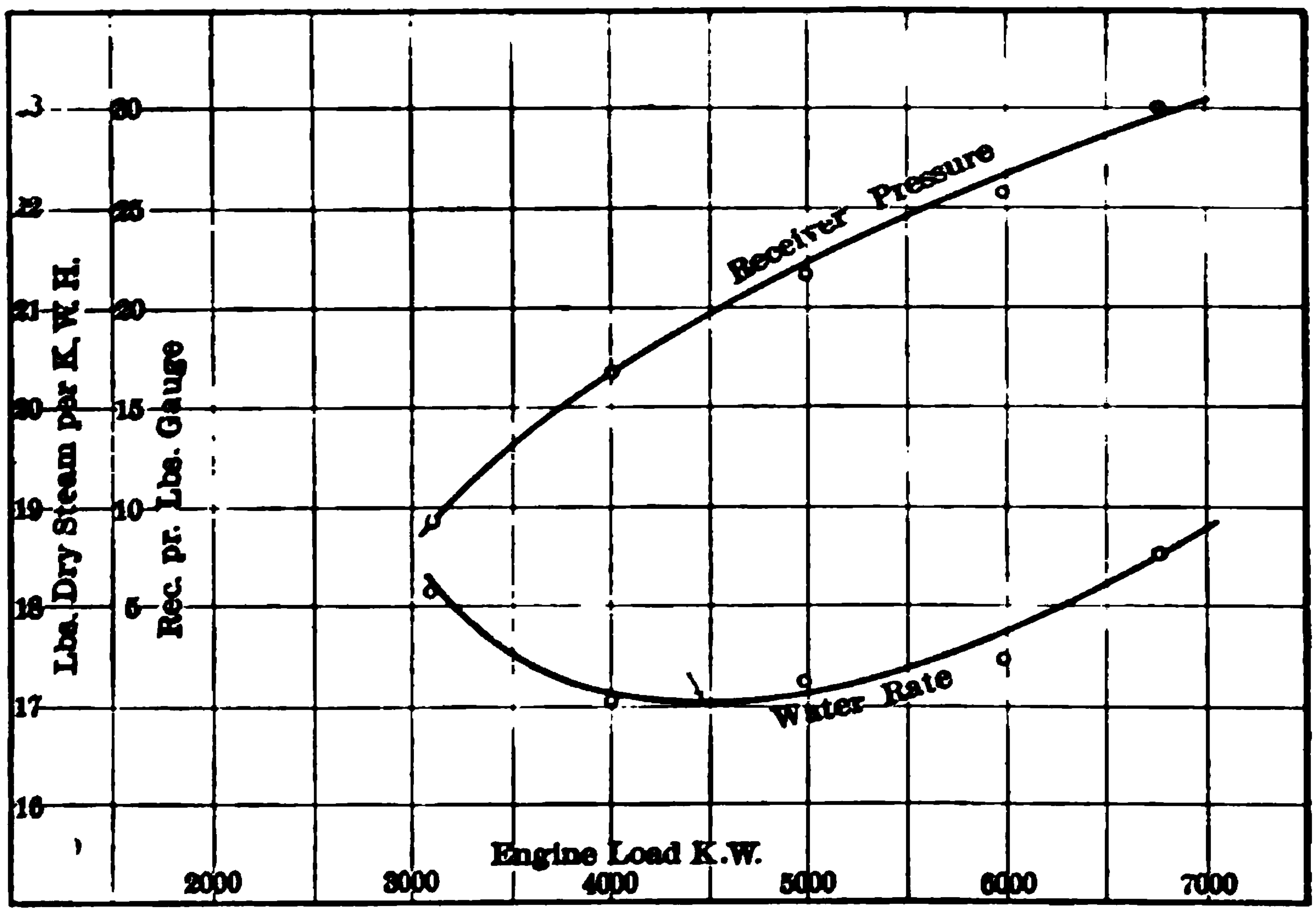


FIG. 194c. Performance of 7500-Kw. Engine at 59th Street Station of Interborough Rapid Transit Company, New York, with Varying Receiver Pressure.

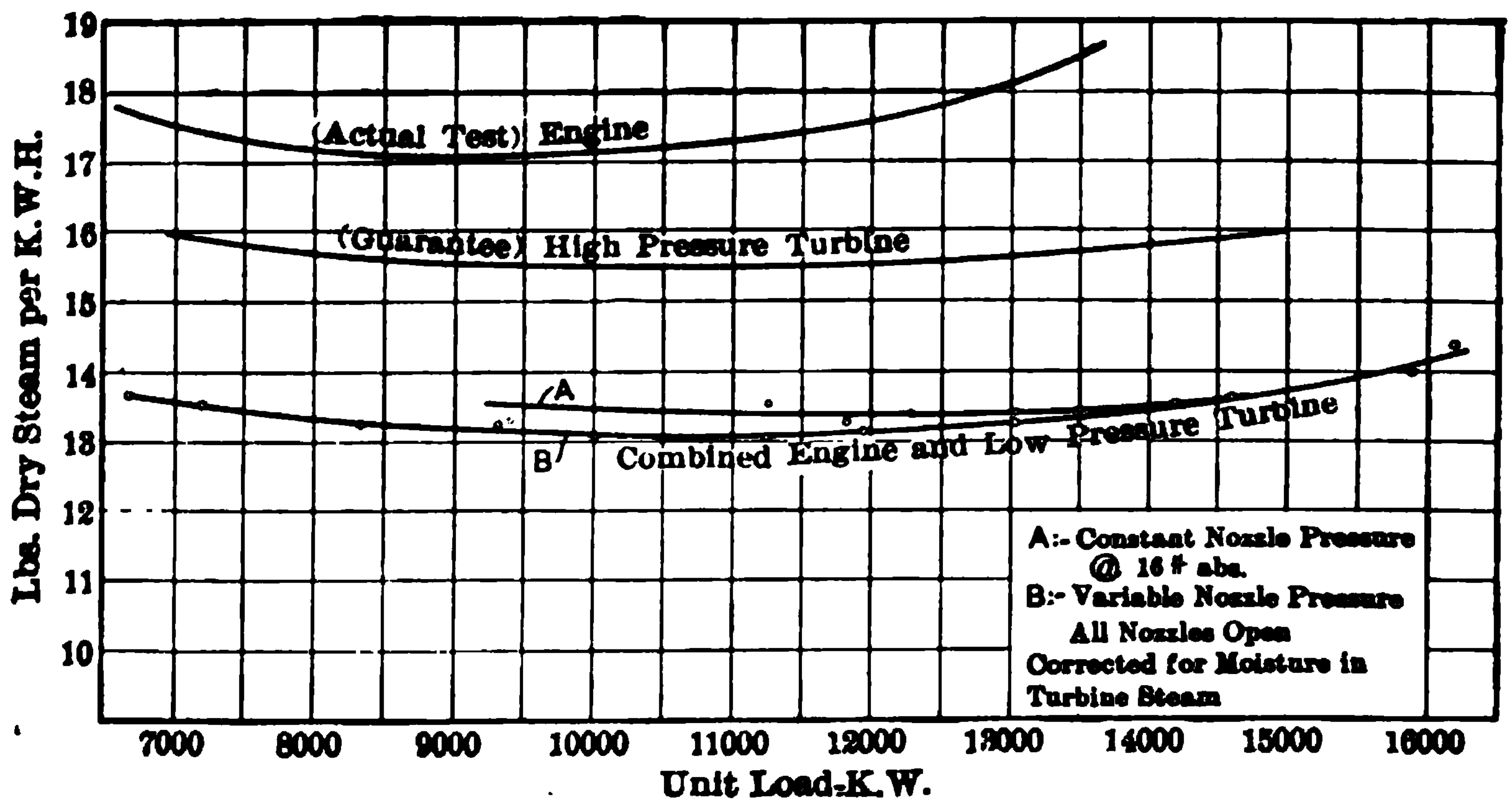


FIG. 194d. Comparison of Economy Curves: 7500-Kw. High-pressure Turbine, 7500-Kw. Engine and Combined Engine and Low-pressure Turbine at the 59th Street Station of the Interborough Rapid Transit Company, New York.

with that of the engine-driven generator. The turbine is equipped with the usual emergency speed limit attachment for cutting off the steam supply should the speed exceed a predetermined limit. The

performance of one set of engines, a high-pressure turbine of the equivalent total capacity, and that of the combined engine and low-pressure turbine, are illustrated in Fig. 194d. The conclusions drawn from an exhaustive series of tests at this station are that the addition of low-pressure turbines effected:

a. An increase of 100 per cent in maximum capacity of plant.

b. An increase of 146 per cent in economic capacity of plant.

c. A saving of approximately 85 per cent of the condensed steam for return to the boiler.

d. An average improvement in economy of 13 per cent over the best high-pressure turbine results.

e. An average improvement in economy of 25 per cent (between the

FIG. 195. Rateau Low-Pressure Steam Turbine Installation.

limits of 7000 kw. and 15,000 kw.) over the results obtained by the engine units alone.

f. An average unit thermal efficiency of 20.6 per cent between the limits of 6500 kw. and 15,500 kw.

Low-pressure turbines are frequently installed in connection with regenerative accumulators, to rolling-mill engines, steam hammers, and other appliances using steam intermittently, and have proved to be paying investments. A typical installation of this character is to be found at the South Chicago Division of the International Harvester Company. The front elevation of the turbine and regenerator installation is shown in Fig. 195 and the general arrangement of the regenerator is shown in Fig. 196. The regenerative accumulator is intended to regulate the intermittent flow of steam before it passes to the turbine.

The steam collects and is condensed as it enters the apparatus and is again vaporized during the time when the exhaust of the engines diminishes or ceases.

The regenerator consists of a cylindrical boiler-steel shell divided into two similar chambers by a central horizontal diaphragm. In each compartment are a number of elliptical tubes A, each of which is perforated with a number of $\frac{3}{4}$-inch holes. The spaces surrounding the tubes and, under certain conditions, the tubes themselves are filled with water to a height of about four inches above the top of the upper tubes. Baffle plate B serves to separate the entrained moisture from the steam. The operation is as follows: Exhaust steam enters the apparatus at N, passes to the interior of the elliptical tubes, and escapes

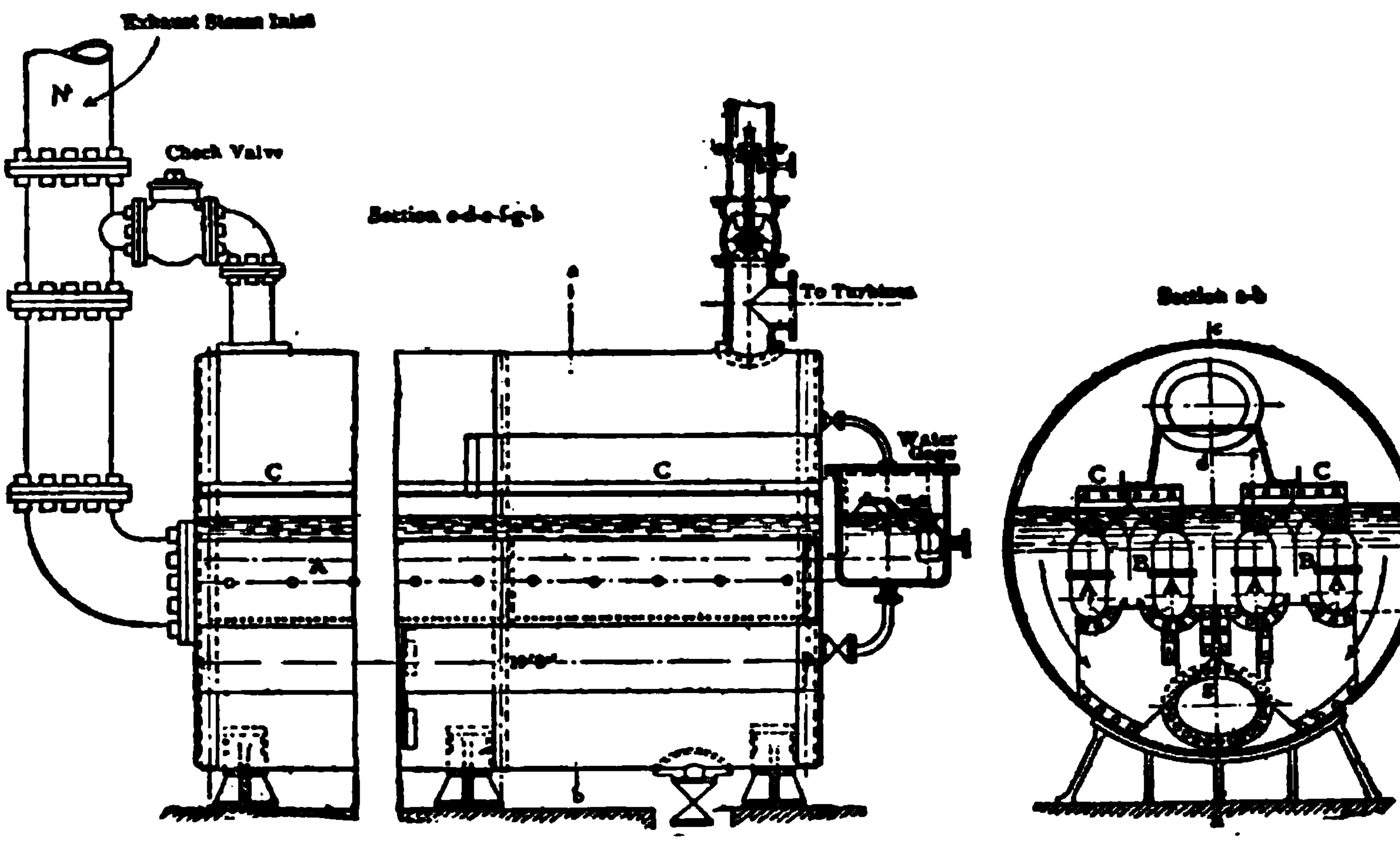


FIG. 196. Rateau Regenerator Accumulator.

into the steam space through the perforations and thence to the turbine. When the supply of steam from the main engine ceases, the pressure in the regenerator decreases, the water liberates part of the heat it has absorbed and a uniform flow of low-pressure steam is given off. The continued demand of the turbine reduces the pressure in the accumulator and causes the steam still retained in the tubes to escape, thereby maintaining the circulation of the water (indicated by arrowheads) and facilitating the liberation of steam. Suitable valves regulate the limits of pressure in the accumulator and prevent the return of water to the main engine. Low-pressure turbines develop one electrical horsepower hour on a steam consumption of about 30 pounds with initial pressure of 15 pounds absolute and a back pressure of 1.5 pounds absolute. Fig. 197 gives the performance of the 500-kilowatt Rateau turbine at the International Harvester Works, South Chicago, l.,

average initial pressure of 16 pounds absolute, condenser pressure 1.5 pounds absolute.

Low-pressure turbines equipped with special expanding nozzles, or the equivalent, to receive steam at high pressure direct from the boilers are known as *mixed pressure* turbines. With this construction the *full power* of the turbine can be developed with (1) all low-pressure steam, (2) all high-pressure steam, (3) any proportion of high and low

Fig. 196a. Typical Double-deck Installation, Fort Wayne and Wabash Valley Traction Company, Spy Run Station.

pressure steam. In the Curtis mixed-pressure turbine this transition from all low pressure to all high pressure, through all the conditions intermediate between these extremes, is provided for automatically by the turbine governor; a deficiency of low-pressure steam causes the high-pressure nozzles to open automatically. With this arrangement it is not necessary for purposes of economy to proportion exactly the low-pressure turbine to the amount of exhaust steam available, but within limits it may be made as large as the load demands.

Low-pressure Turbines: Power & Engr., July 6, 1909, p. 1, Nov. 30, 1909, p. 905; Prac. Engr. U. S., Mar. 1, 1909, p. 169; Eng. Mag., Apr. and May, 1907; Iron Age, Jan. 7, 1909.

195. Advantages of the Steam Turbine. — The principal advantages of the steam turbine are (1) simplicity; (2) economy of space and foundation; (3) absence of oil in condensed steam; (4) freedom from vibration; (5) uniform angular velocity; (6) large overload capacity; and (7) high efficiencies for large variations in load. The reciprocating engine is well adapted for pumping stations, direct-current generators, compressor plants, hoisting engines, and the like, requiring low angular velocity, but its place is being rapidly taken by the steam turbine for alternating-current dynamos, centrifugal pumps and blowers, requiring high angular velocity.

FIG. 197.

196. Simplicity. — Although composed of a large number of parts as compared with a reciprocating engine of the same capacity, there are few moving parts and rubbing surfaces. The only contact between rotor and stator is in the main bearings, and the problem of lubrication is therefore a simple one. The absence of pistons, stuffing boxes, dash pots, etc., reduces the cost of maintenance and attendance to a minimum and limits the possibility of leakage.

197. Economy of Space and Foundation. — Fig. 198 (J. R. Bibbins, *Power*, January, 1905) shows the relative floor space required for different types of slow-speed reciprocating engines and Westinghouse-Parsons steam turbines. The floor space required by the Curtis turbine is considerably less than for the Parsons type, as it is vertical with the generator mounted above the turbine. Vertical three-cylinder compound

Corliss engines of the New York Edison type require the least floor space of any large slow-speed reciprocating engines, but take up about twice the space of a Parsons turbine installation of the same size. With high-speed engines of the Willans central-valve type the comparative economy in space is less marked. The curves refer to the space occupied by engine and generator alone, whereas in a modern turbine installation with surface condenser the condenser equipment may occupy as much space as, if not more than, the turbine proper, and considering the small space taken up by the barometric condenser, such as is used in connection with the New York Edison engines, the economy is not so evident. In recent turbine installations the baro-

Fig. 196.

metric condenser is finding much favor, and in such instances the curves may be taken to indicate the relative floor space for the entire equipment of prime mover and auxiliaries.

The weight of the steam turbine is very small compared with a reciprocating engine of the same horse power. The New York Edison engines and generators weigh more than eight times as much as a turbine installation of equal capacity. The turbine for this reason, and also because of the total absence of vibration, requires a relatively light foundation. In many instances the foundation consists of steel beams with concrete arches sprung between them resting upon the floor, and the basement underneath may be used for the condenser instead of the massive foundation required for the reciprocating engine.

198. Absence of Oil in Condensed Steam. — As the steam turbine requires no internal lubrication, oil does not come in contact with the steam, and the condensed steam from the surface condensers is available for boiler-feeding purposes without purification. In many cases the re-use of condensed steam effects a large saving in cost of feed water and in expense for maintenance and cleaning of boilers. The amount of entrained air is reduced to a minimum and consequently the work of the air pumps lessened.

199. Regulation. — The variable pressure at the crank pin of a reciprocating engine necessitates the use of a heavy fly wheel to keep the instantaneous angular fluctuation within practical limits. In the steam turbine the motion is purely rotary and a fly wheel is not necessary. In the former there are always instantaneous variations in velocity during each revolution, even with constant load, while in the latter the speed is practically constant. A number of published tests of Parsons and Curtis turbines show an average fluctuation of 2 per cent from no load to full load and 3 per cent from no load to 100 per cent overload. Although closer regulation than this is possible, it is not deemed necessary, particularly in alternating-current work where a comparatively wide range is desirable for parallel operation.

200. Overload Capacity. — A particular advantage of the turbine over the reciprocating engine lies in its greater overload capacity and higher economy at overloads. The maximum economy of the average reciprocating engine lies between 0.75 and full load, whereas the turbine reaches its maximum at about 25 per cent overload. Thus a single turbine unit may economically take the place of two or more reciprocating units for a variable load. A turbine may be readily operated at 100 per cent overload, while the ordinary engine reaches its maximum capacity at about 50 per cent overload. In central lighting and power stations where there are one or more sharp peak loads of short duration, this extreme overload capacity is of marked importance.

201. Efficiency and Economy. — As far as steam consumption is concerned there is practically no difference between the performance of a high-grade piston engine and that of a first-class turbine for sizes under 2000 kw., the choice depending more upon rotative speed, overload capacity and space requirements than upon the heat economy. For sizes over 2000 kw. the fuel consumption lies in favor of the turbine. A comparison of Fig. 148, showing typical economy curves of high-speed single-valve, non-condensing engines, and of Fig. 198c, showing similar curves for small non-condensing turbines, is somewhat

in favor of the piston engine, though the difference is small; whereas a comparison of the turbine and engine curves in Fig. 194d, showing the performance of very large units, is decidedly in favor of the turbine.

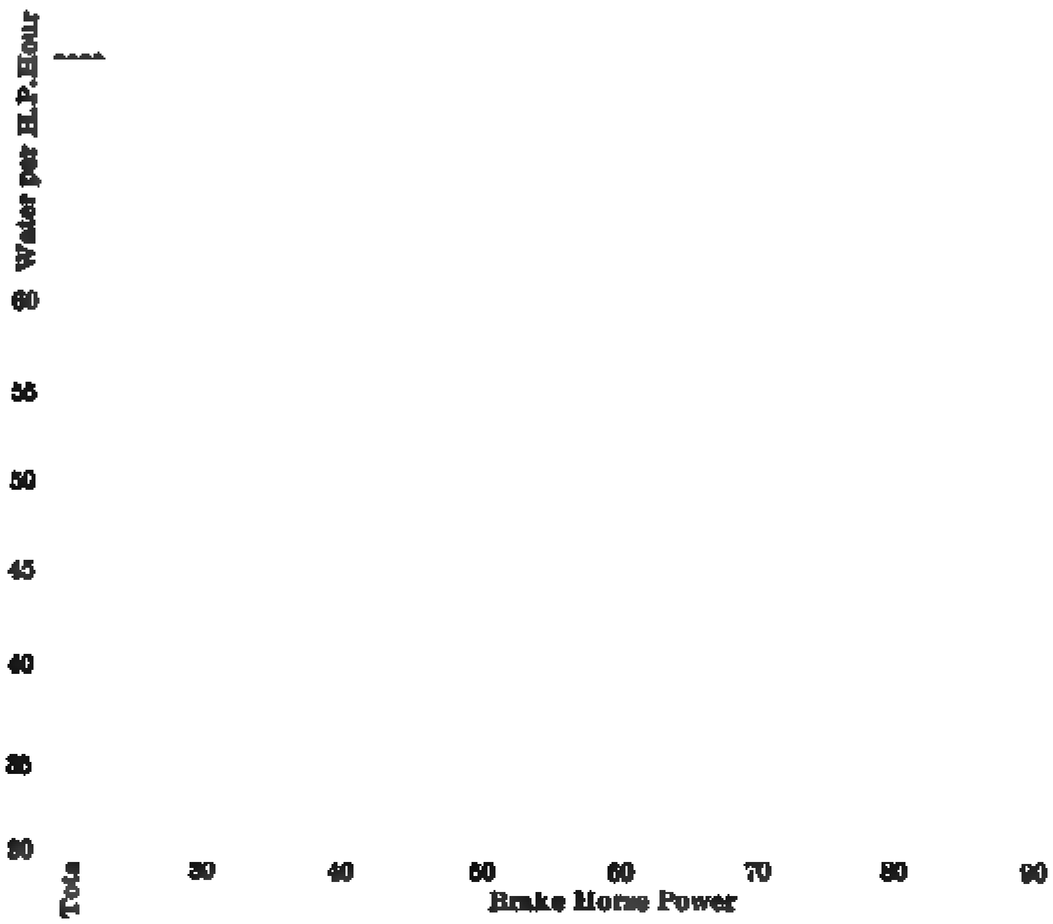


FIG. 198a. Typical Performance of a 90 H.P. Terry Steam Turbine.

Any number of individual tests may be cited showing superiority in fuel consumption of the piston engine over that of a turbine of equivalent capacity and vice versa, but when the machines are designed for

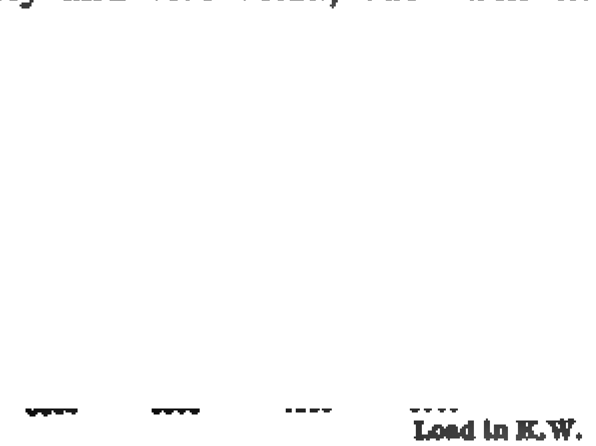


FIG. 198b. Typical Performance of 9000 Kw. Curtis Turbine; 200 Lbs. Gauge Pressure, 125° Superheat, 29 Inches Vacuum.

the same operating conditions the results are practically the same for all sizes under 2000 kw. Tables 39 to 43 give the general condition of operation and the steam consumption of exceptionally good piston engines of various sizes and types, and Table 48 similar data of first-

class turbines. A study of these tables will show that the choice must be based on other factors than the steam consumption. In a general sense, the piston engine is superior to the turbine for high back pressures, slow rotative speeds and heavy starting torques, while the turbine has practically superseded the engine for large central station units and for auxiliaries requiring high rotative speed. Recent tests of the Melville reduction gear (*Machinery*, Feb., 1910) show exceptionally high efficiencies for sizes as large as 6000 kilowatts, and it is not

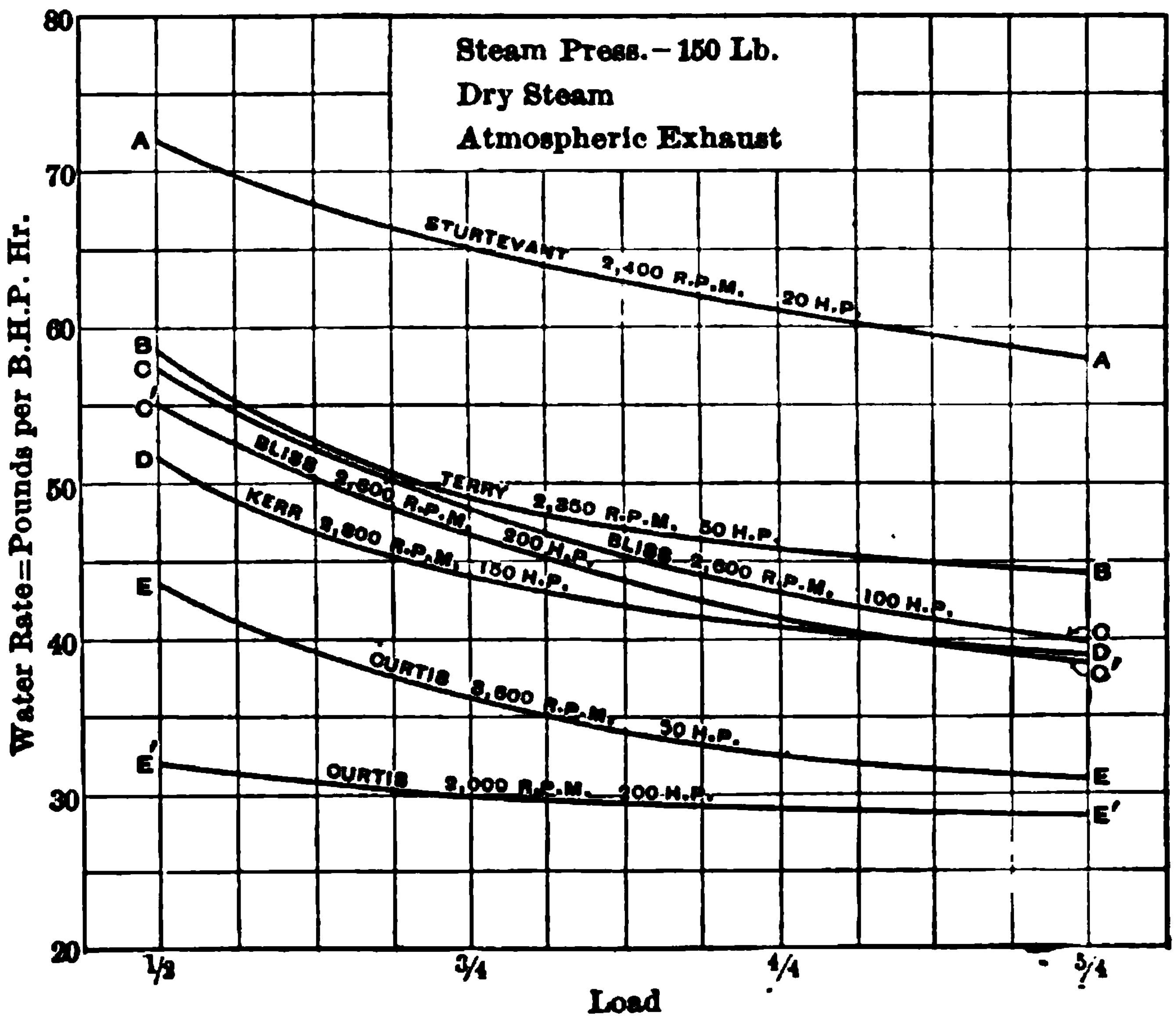


FIG. 198c.

unlikely that the turbine equipped with this device will offset the low rotative speed factor of the piston engine.

If the tests of steam turbines and piston engine could be made at some standard initial pressure, back pressure and quality or superheat, then a comparison could readily be made, but both types of prime movers are designed to give the best results for special operating conditions, and any marked departure from these conditions will result in loss of economy. It is frequently desired, however, to make a comparison between the economy of the different machines, and the following methods are in vogue:

(1) Steam consumption under assumed conditions.

(2) Heat consumption per horse-power per minute above the ideal feed-water temperature.

(3) Efficiency ratio or ratio of ideal to actual.

Standard Correction Curves.

This method for comparing engines or turbines or both is best illustrated by a specific example: Suppose it is required to compare the full-load performance of a 125-kw. direct-connected piston engine with that of a 125-kw. turbo-generator with operating conditions as follows:

	Steam Consumption, Lbs. per Kw.-Hour.	Initial Pressure, Lbs. Absolute.	Vacuum, Inches of Hg.	Superheat, Deg. F.
Engine	25.0	160	25.5	0
Turbine	22.7	110	28.0	125

Manufacturers of steam turbines have provided correction curves as illustrated in Fig. 194d, showing the influence of varying vacuum, superheat and pressures on the steam consumption.* From curve *B*, we find that the steam consumption of the turbine should be

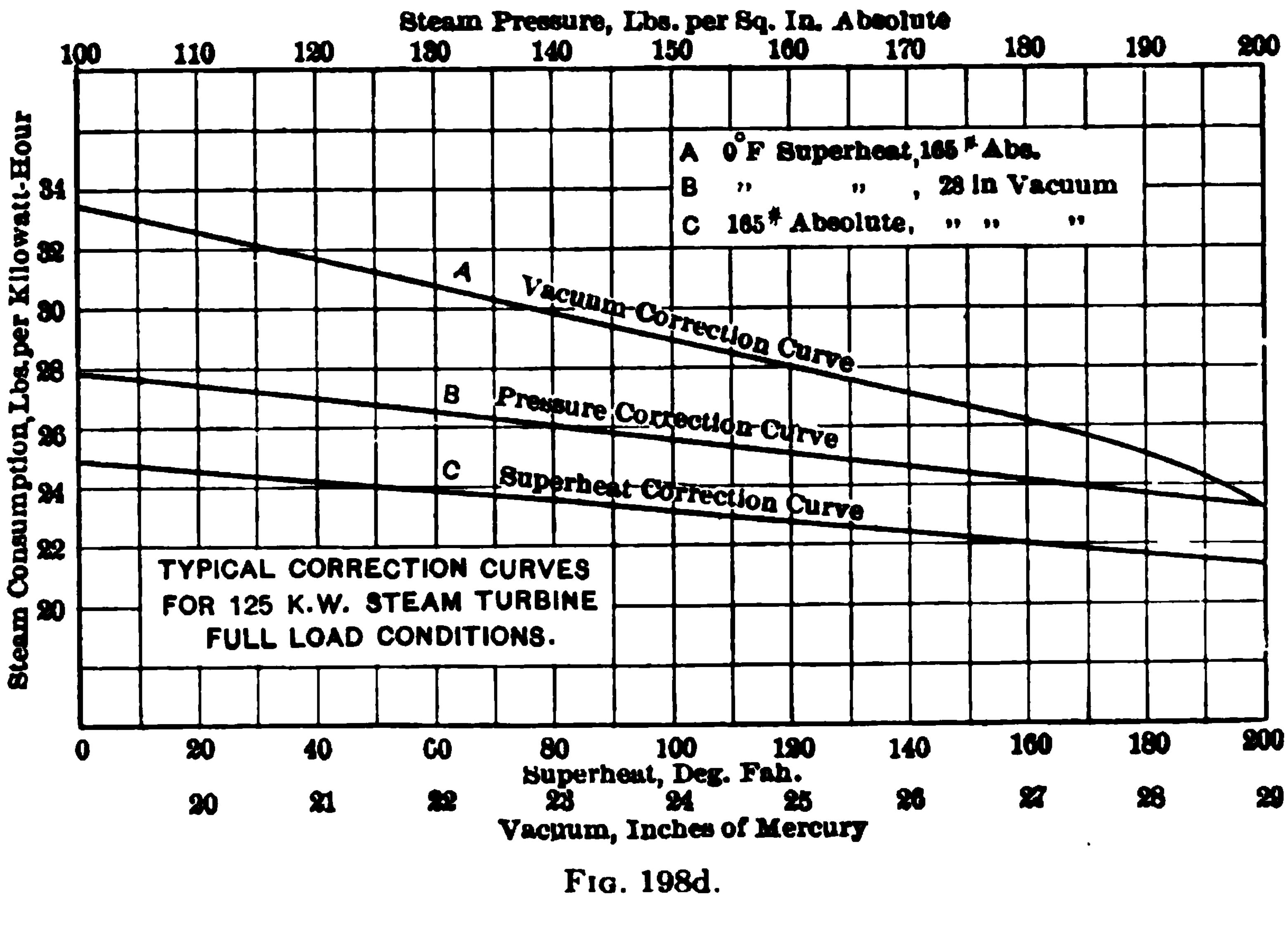


FIG. 198d.

decreased 2.5 pounds to give the equivalent at 160 pounds initial pressure; from curve *A* it should be increased 2.5 pounds to give the equivalent at 25.5 inches of vacuum, and from curve *C* it should be increased 2.5 pounds to give the equivalent at 0 degree superheat. The full-load steam consumption for the turbine under the engine conditions is therefore 22.7 − 2.5 + 2.5 + 2.5 = 25.2 pounds per kw.-hour.

* These curves are drawn to a much larger scale than the reproduction given here.

The *ratio* method is also used in this connection, thus: The full-load steam consumption at 160 pounds pressure, curve *B*, Fig. 194d, is multiplied by the ratio $\frac{25}{27.5}$ to give the equivalent consumption at 110 pounds (25 is the steam consumption at 160 pounds and 27.5 the consumption at 110 pounds). Similarly the correction ratio to change the consumption at 28 inches of vacuum to 25.5 is $\frac{25.5}{25}$, and to correct 125° F. superheat to 0° F. is $\frac{25}{22.5}$.

SUMMARY.

$$\text{Pressure correction } \frac{25}{27.5} = 0.91 = -\ 9\%.$$

$$\text{Vacuum correction } \frac{27.5}{25} = 1.10 = 10\%.$$

$$\text{Superheat correction } \frac{25}{22.5} = 1.11 = 11\%.$$

Net correction 12%.

Corrected steam consumption = 22.7 + 0.12 × 22.7 = 25.4 pounds per kw.-hr.

The ratio method is generally used if the difference between the corrected steam consumption and that of the correction curves for the same conditions is greater than 5 per cent ("The Steam Turbine," Moyer, p. 128).

This ratio method for correcting steam consumption at full load may be used without appreciable error for half to one and one half load and is the only practical method for quarter load (*Engrg.* London, March 2, 1906).

Heat Consumption.

The heat consumption B.T.U. per unit output per minute above the ideal feed water temperature may be expressed

$$\frac{W(H_1 - q_2)}{60}. \quad \text{See equation (75b).}$$

For the case cited above

$$\text{Engine,} \qquad \frac{25\ (1194.1 - 98)}{60} = 455 \text{ B.T.U.}$$

$$\text{Turbine,} \qquad \frac{22.7\ (1264.2 - 70)}{60} = 451 \text{ B.T.U.}$$

Efficiency Ratio.

The efficiency ratio, or the extent to which the theoretical possibilities are realized, may be expressed

$$E_r = \frac{2545}{W\,(H_1 - H_2)}. \quad \text{See equation (76b).}$$

For the case cited above

$$\text{Engine,} \qquad \frac{2545}{25\,(1194.1 - 915)} = 0.366.$$

$$\text{Turbine,} \qquad \frac{2545}{22.7\,(1264.2 - 915.3)} = 0.322.$$

In the assumed case the turbine is the more economical in heat consumption, but the engine is the more perfect of the two as far as theoretical possibilities are concerned.

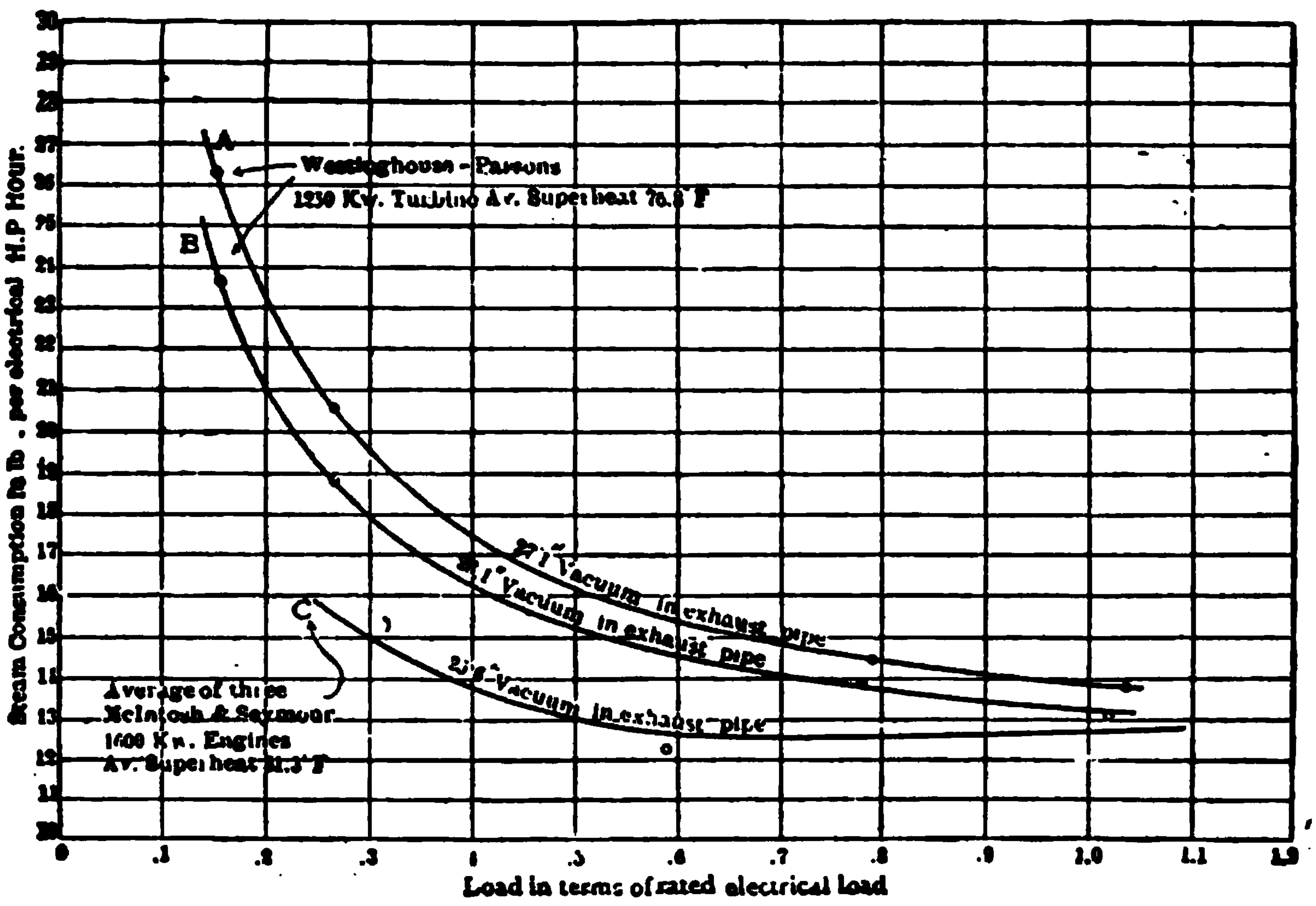


FIG. 199.

202. First Cost. — Steam turbines, generally speaking, are about 10 per cent lower in first cost than high-grade compound engines of equivalent power. The following table gives an idea how the price varies with the conditions of operation. The figures are approximate only and refer to the cost of the turbine and generator exclusive of auxiliaries.

TABLE 48.—EXAMPLES OF STEAM TURBINE ECONOMY.

Index.	Kind of Turbine.	Rated Capacity.	References.	Load.	Per Cent of Rated Load.	Initial Press., Lbs. Gauge.	R.P.M.	Superheat at Throttle, Deg. F.	Back Press., Lbs. Abs.	Lbs. of Steam per Kw. Hr.	Lbs. of Steam per B.H.P. Hr.	B.T.U. per Elec. H.P. per Min.	B.T.U. per Brake H.P. per Min.
0	Curtis	9000 kw.	Proc. Nat. Elec. Light Assn., June 4, 1907.	10186 kw.	113	176	750	147	0.26	12.9	......	200	....
1	....do	5000 kw.	Engr. U.S., July 2, 1906, p. 444	5000 kw.	100	175	650	150	1.20	16.07	......	241	....
2	....do	2000 kw.	....	2023 kw.	101	166.3	900	207	1.10	15.02	......	231	....
3	....do	500 kw.	G. H. Barrus, Newport Station, Old Colony Ry. Co.	529 kw.	106	144.6	1812	0	0.50	19.78	......	280	....
4	....do	500 kw.	....do	528.9 kw.	105	147	1835	150.5	0.55	17.79	......	271	....
5	....do	500 kw.	....do	519.7 kw.	103	145	1850	289.6	0.45	15.91	......	258	....
6	....do	100 kw.	St. Ry. Jour., Oct. 15, 1904, p. 707.	100 kw.	100	125	3600	0	1.20	25.70	......	353	....
7	....do	75 kw.	....do	75 kw.	100	150	2400	0	Atm.	47.00	......	593	....
8	....do	25 kw.	....do	25 kw.	100	150	3600	0	..do..	54.00	......	680	....
9	....do	15 kw.	....do	15 kw.	100	200	4000	0	..do..	57.00	......	722	....
10	De Laval	300 H.P.	Eng. Record, Aug. 2, 1902	352 H.P.	117	198	750	84	1.40	......	13.94	....	272
11	....do	300 H.P.	....do	333 H.P.	111	196.9	747	0	1.70	......	15.17	....	274
12	....do	150 H.P.	Am. Elecn., Aug., 1905, p. 445	160.4 H.P.	106	146	1200	0	1.75	26.49	17.61	....	324
13	....do	55 H.P.	Engr. U.S., June, 1904, p. 418.	59.2 H.P.	107	189	2050	0	2.60	......	24.20	....	430
14	....do	30 H.P.	Trans. A.S.M.E., Vol. 25, p. 1068	30 H.P.	100	125	....	0	2.70	......	22.50	....	406

15	De Laval..........	10 k.w.	Trans. A.S.M.E., Vol. 25, p. 1068.	10 H.P.	100	140	....	0	Atm.	65.00	......	320	....
16	Westinghouse-Parsons	7500 k.w.	Power, Nov., 1907, p. 772.....	9830 kw.	133	177.5	750	95.7	1.30	15.15	......	221	....
17	....do..............	1250 kw.	Elec. World, Feb. 20, 1904, p. 357.	1261 H.P.	....	150.7	....	0	Atm.	......	24.05	....	320
18	....do..............	....do...	....do....................	1346 kw.	109	146	1197	0	1.00	18.79	......	262	....
19	....do..............	....do...	....do....................	1274 kw.	102	146	1199	75	1.00	17.66	......	256	....
20	....do..............	....do...	....do....................	1275 kw.	102	148.6	1201	100	1.00	17.35	......	254	....
21	....do..............	750 kw.	St. Ry. Jour., Oct. 15, 1904, p. 710.	1074 H.P.	....	190	....	0	Atm.	......	28.36	....	481
22	....do..............	....do...	....do....................	1345 H.P.	....	150	1807	152.2	1.00	......	11.42	....	230
23	....do..............	....do...	....do....................	1554 H.P.	....	151	1787	99.7	1.10	......	12.99	....	254
24	....do..............	400 kw.	Elec. World, June 18, 1904, p. 1162.	....	102	154	3545	0	1.60	......	13.91	....	257
25	....do..............	....do...	....do....................	....	102	156	3546	100	1.50	......	12.41	....	242
26	....do..............	....do...	....do....................	....	102	154	3542	180	1.45	......	11.45	....	232
27	Parsons............	3000 kw.	Stodola, "Steam Turbines," p. 286.	2985 kw.	099	155.8	1360	225	1.50	14.77	......	227	....
28	....do..............	1000 kw.	Engng., Sept. 6, 1901, p. 339...	1190 kw.	119	129.5	1487	18.4	0.90	19.42	......	274	....
29	....do..............	500 kw.	....do....................	515 kw.	103	145.9	2500	70	1.42	21.34	......	306	....
30	....do..............	200 kw.	....do....................	204 kw.	102	119	3000	0	1.52	24.21	......	333	....
31	....do..............	100 kw.	....do....................	108.5 kw.	108	127	4800	0	1.48	26.81	......	370	....
32	....do..............	50 kw.	....do....................	52.7 kw.	105	126	5044	0	0.98	28.00	......	388	....
33	....do..............	24 kw.	....do....................	24.7 kw.	103	79.9	4990	0	0.59	28.79	......	403	....

APPROXIMATE COST OF STEAM TURBINES AND GENERATORS.

In Dollars per Kilowatt. Rated Capacity.

	Kilowatts.										
	25	50	75	100	200	300	500	1000	2000	4000	6000
Direct current:											
Non-condensing	55	47	42	38	32	32	....	....	......	......	......
Condensing	60	51	46	43	36	36	....	....	......	......	......
Alternating current:											
25 cycles	....	....	....	....	....	35	32	28	25	21	20
60 cycles	....	....	....	....	....	35	30	27	25	21	20

203. Cost of Operation. — Data pertaining to the cost of operating steam-turbine and reciprocating engine plants and combinations of both will be found in Chapter XVII. The following table, contributed by H. G. Stott, Superintendent Motive Power of the Interborough Rapid Transit Company, New York, gives an excellent comparison of the *relative* maintenance and operating costs (March, 1910) of the three types of steam power plants as applied to large central stations for electric street railways.

RELATIVE COSTS PER KILOWATT-HOUR. DISTRIBUTION OF MAINTENANCE AND OPERATION.

	Reciprocating Steam Plant.	Steam Turbine Plant.	Reciprocating Engines and Low-pressure Steam Turbines.
Maintenance.			
1. Engine room, mechanical	2.59	0.51	1.55
2. Boiler or producer room	4.65	4.33	3.55
3. Coal and ash handling apparatus	0.58	0.54	0.44
4. Electrical apparatus	1.13	1.13	1.13
Operation.			
5. Coal	61.70	55.53	46.48
6. Water	7.20	0.65	0.61
7. Engine room labor	6.75	1.36	4.06
8. Boiler or producer room labor	7.20	6.74	5.50
9. Coal and ash handling labor	2.28	2.13	1.75
10. Ash removal	1.07	0.95	0.81
11. Electrical labor	2.54	2.54	2.54
12. Engine room lubrication	1.78	0.35	1.02
13. Engine room waste, etc.	0.30	0.30	0.30
14. Boiler room lubrication, etc.	0.17	0.17	0.17
Relative operating cost, per cent	100.00	77.23	69.91
Relative investment, per cent	100.00	75.00	80.00
Probable average cost per kw	125.00	93.75	100.00
Probable fixed charges	11%	11%	11%

For steam turbine plants larger than 60,000 kw. the cost per kilowatt may be reduced to $75.00.

204. Influence of Superheat. — The use of superheated steam increases the economy of the reciprocating engine about 1 per cent for every 10 to 20 degrees of superheat, depending upon the conditions of operation, the gain being due mainly to the reduction of cylinder condensation. Cylinder condensation is reduced not only because of the excess heat available for the evaporation of moisture but also because superheated steam has a lower conductivity than wet steam, and less heat is given up to the cylinder walls for the same difference of temperature. In the steam turbine this difference of temperature is much smaller, since high- and low-pressure steam do not alternately come in

Fig. 200.

contact with the same surface as is the case with the reciprocating engine, and the time of contact is considerably less, due to the comparatively high velocities. With a well-lagged casing, therefore, the condensation due to this cause is insignificant compared with that of the reciprocating engine, and the beneficial effect of superheat is much more pronounced. Friction of the steam, which in the reciprocating engine is negligible, and which may be a source of considerable loss in the turbine, is greatly reduced by the use of superheated steam, as is also the " windage " loss due to the rapid revolution of the wheels.

The problem of cylinder lubrication is sometimes a difficult one in steam engines using a high degree of superheat, and trouble is frequently experienced due to the unequal expansion of the metal. In

the steam turbine the latter difficulty is not so pronounced and no internal lubrication is necessary, hence higher degrees of superheat are permissible. For maximum economy the steam at the end of expansion should be free from moisture. Assuming purely adiabatic expansion, the steam in expanding from 165 pounds to 1 pound absolute would have to be superheated about 500 degrees F., giving the steam an actual temperature of 800 degrees F. A study of some 100 tests made in this country gives about 250 degrees superheat as a maxi-

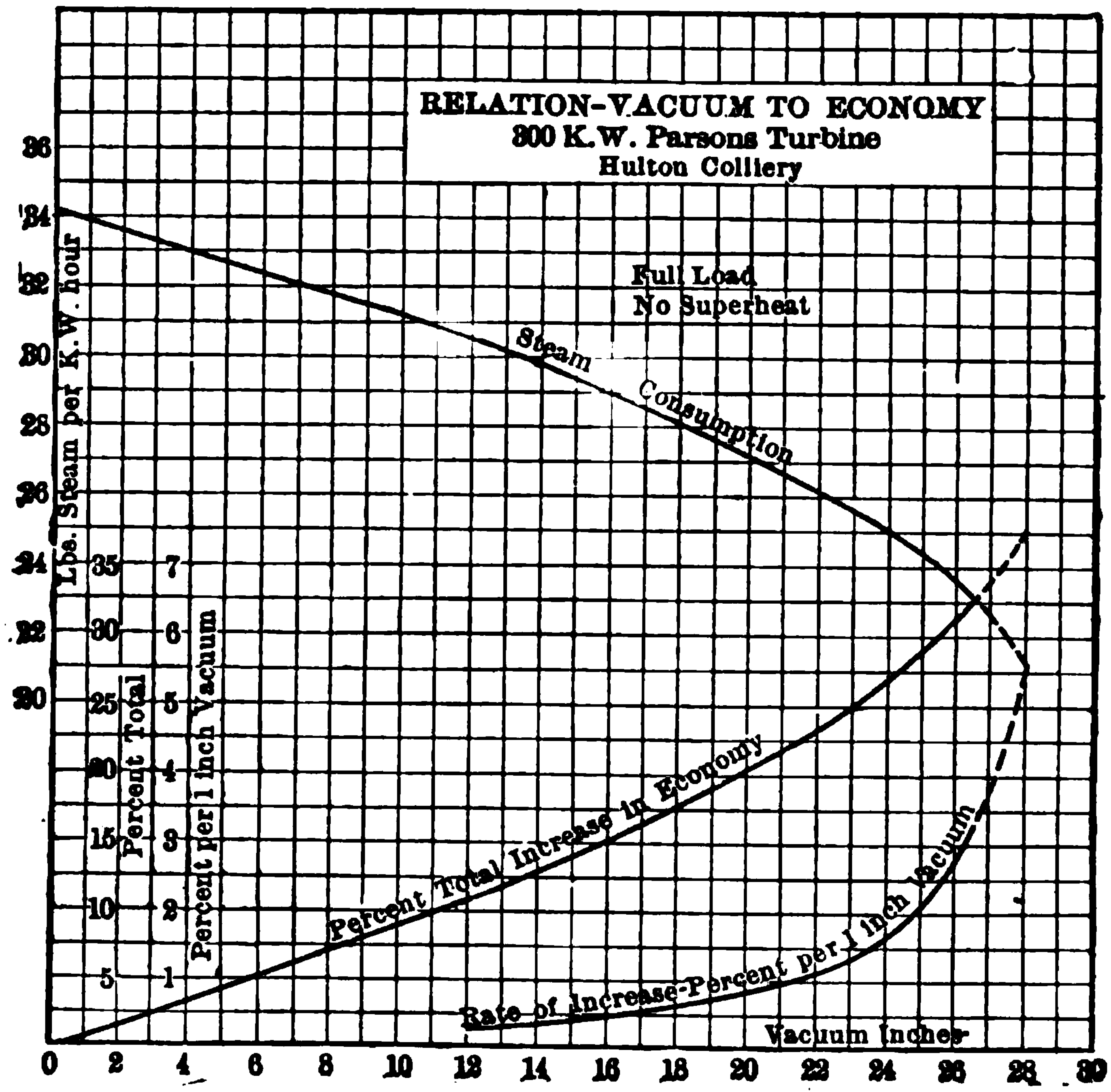


FIG. 201.

mum and 100 degrees to 150 degrees F. as an average. In Europe reciprocating engines are operating with superheat as high as 350 degrees F. and turbines 300 degrees F. The additional fixed and operating costs of superheating must be considered in determining the net gain, since the decrease in steam consumption does not represent the actual saving. With pressures of 175 pounds gauge or less, and not to exceed 200 degrees F. superheat, the net gain has in most cases proved a substantial one. With higher temperatures and pressures the cost of maintaining the superheat may increase more rapidly than the saving in steam consumption, until a limit is reached beyond which no

advantage is gained. The relation between superheat and steam consumption for a 400-kilowatt Westinghouse-Parsons turbine is illustrated in Fig. 200. Fig. 202 gives a similar comparison for a 1500-kilowatt turbine. (J. R. Bibbins, *Power*, January, 1905.)

205. Influence of High Vacua. — The possible economy of the reciprocating engine is greatly restricted by its limited range of expansion. Cylinders cannot be profitably designed to accommodate the rapid increase in the volume of steam when expanded to very low pressures. For example, the specific volume of 1 pound of steam under a vacuum of 29 inches (referred to a 30-inch barometer) is about 650 cubic feet, or nearly double its volume under a vacuum of 28 inches. Usually the exhaust is opened at a pressure of 6 or 8 pounds absolute and consequently a large proportion of the available energy is lost. The lower vacuum in the exhaust pipe, therefore, serves only to diminish the back pressure and does not affect the completeness of expansion. Even if it were practical to expand to 1 pound absolute, the increased condensation in the reciprocating engine would offset any gain due to expansion unless the steam were highly superheated. A study of a number of tests of reciprocating engines shows a slight improvement due to increasing the vacuum beyond 26 inches. Tests of steam turbines show a decrease of 3 to 4 per cent in steam consumption for each inch increase of vacuum between 25 and 29 inches, for with a well-lagged casing cylinder condensation is practically absent, since the high- and low-temperature steam do not alternately come in contact with the metallic surfaces as is the case with the reciprocating engine (Figs. 201 and 201a). Fig. 230 shows a relation between the power consumption of the auxiliaries and the total output of the station at different loads for a Parsons steam turbine installation and Fig. 231 shows a similar relation for the 2000-kilowatt Curtis turbine. The power consumption in the latter case is higher on account of the high temperature of cooling water. Table 55 gives the power required for the auxiliaries in a number of stations. A high vacuum may be limited by the initial temperature of the cooling water. The difference in temperature between

Fig. 201a.

inlet and discharge should be greater than 10 degrees, since otherwise the amount of circulating water per pound of steam becomes excessive and increases the work of the pumps. For example, the temperature of steam corresponding to a vacuum of 28 inches or 1 pound absolute is 102 degrees, and with cooling water at 75 degrees F., and the discharge at 95 degrees F., the *theoretical* ratio of cooling water to steam necessary to maintain this vacuum will be about 50 and the *actual* nearer 70. From Table 50 it will be seen that a 28-inch vacuum referred to a 30-inch barometer is obtained with an average ratio of 50 pounds of cooling water at 70 degrees F. per pound of steam. The cost of high-vacuum apparatus is not proportional to the vacuum, but increases much more rapidly, as shown in Fig. 232. These estimates show averages and not specific costs.

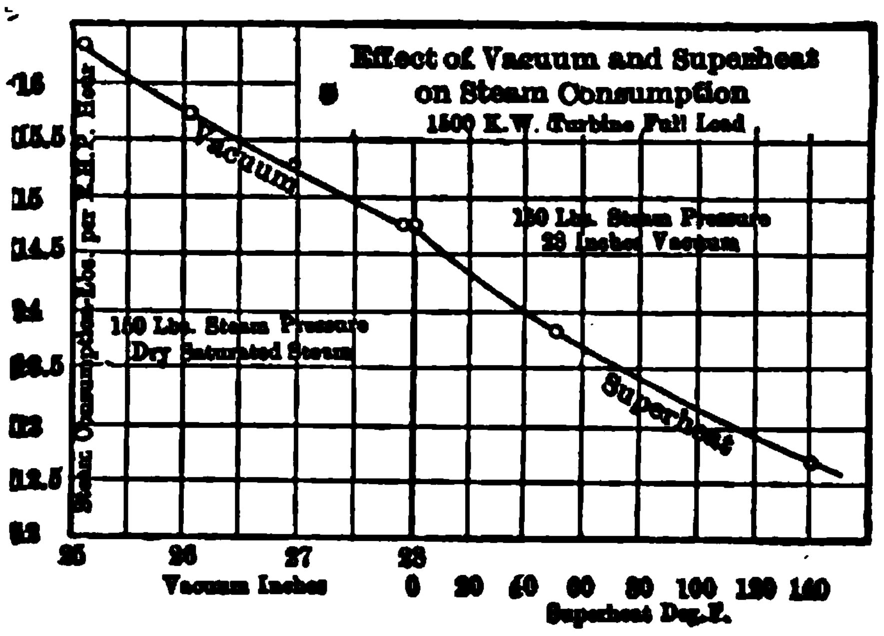


Fig. 202.

Fig. 202 shows the effect of superheat and vacuum on the economy of a 1500-kilowatt Westinghouse-Parsons turbine. Figs. 200 to 202 are taken from "Steam Power Plants," by J. R. Bibbins, as published in *Power*, January, 1905.

Reciprocating Engine vs. Turbine: Power, April, 1904, p. 232, May, 1904, p. 298; Engr. U. S., Nov. 1, 1905, p. 711; Elec. World, April 2, 1904, p. 651; Eng. Mag., Sept., 1905, p. 935; Power, Feb., 1906, p. 83; Elec. Age, June, 1905, p. 478; Elec. Rev., Dec. 23, 1904.

Steam Turbine Design: Eng. Rec., July 22, 1905, p. 101; St. Ry. Jour., Dec. 20, 1902, p. 988; Engr., Lond., Jan. 8, 1904, p. 34, May 13, 1904, p. 481, Dec. 27, 1907; Electrician, Lond., March 24, 1905; Power, Dec., 1905; Mech. Eng., Feb. 7, 1908; Engineering, Dec. 13, 1907.

Theory and Design of Steam Turbines: Engr. U. S., Dec. 16, 1907, p. 1126 (serial), March 15, p. 201; Revue de Mécanique, Oct. 31, 1907; Engr., Lond., Oct. 4, 1907; Eng. Rev., May, 1904; Mech. Engr., Oct. 28, 1905; Eng. Rec., July 22, 1905, p. 101, May 7, 1904, p. 581.

Modern Steam Turbine Plants: Power, Dec., 1906, p. 717, Dec., 1907; Engr. U.S., Nov. 15, 1906, p. 733, March 15, 1907, p. 304; St. Ry. Jour., Oct. 19, 1907; Elec. World, July 22, 1905, Feb. 15, 1908; Eng. Rec., March 4, 1905.

Tests of Westinghouse-Parsons Turbines: Engr. U.S., Dec. 1, 1904, p. 802; St. Ry. Jour., Dec. 19, 1903, p. 1063; Power, March, 1904, p. 127, April, 1904, p. 239, Aug., 1905, p. 466; Elec. World, Sept. 6, 1902, p. 360; Eng. Rec., July 29, 1905, p. 134.

Governing Steam Turbines: Harvard Engineering Jour., 1908.

A Recent Comparison of Turbines and Engines: Eng. Rec., Feb. 19, 1910.

Internal Losses of a Steam Turbine: Power & Engr., Aug. 24, 1909.

The Principles of Steam-turbine Buckets: Power, Mar. 17, 1908.

CHAPTER XI.

CONDENSERS.

206. General. — A pound of dry steam at atmospheric pressure (30 inches mercury) occupies a volume of 26.8 cubic feet. Suppose these 26.8 cubic feet of steam were contained in a closed vessel, and that the steam was subsequently condensed and its temperature lowered by suitable means to say 110 degrees F. The condensed steam would occupy only about $\frac{1}{1700}$ of its original volume, and the pressure would fall to 2.6 inches of mercury, the latter pressure being due to the tension of the aqueous vapor at the given temperature. That is to say, the best vacuum theoretically attainable under the given conditions would be $30 - 2.6 = 27.4$ inches. The lower the temperature to which the condensed steam is reduced the more nearly perfect will be the vacuum attained.

If air is mixed with the steam the vacuum will be still more imperfect. Thus, suppose the vessel to contain one pound of steam and one-tenth of a pound of air under atmospheric pressure. The volume of the closed vessel in this case must be $26.8 + 1.69 = 28.49$ cubic feet.

After the steam has been condensed and its temperature reduced to 110 degrees F. the remaining pressure will be due to the aqueous vapor tension plus the pressure due to the air, or $2.6 + 1.51 = 4.11$ inches mercury, and the maximum vacuum attainable will be 25.89 inches. In practice air is always present in exhaust steam. A condenser is a device in which the process of condensation and subsequent removal of the air and condensed steam is continuous, the degree of vacuum obtained depending upon the tightness of valves and joints, the quantity of entrained air, and the temperature to which the condensed steam is reduced.*

The degree of vacuum may be expressed in different ways. (1) Excess of the atmospheric pressure over the observed vacuum. For example, a 26-inch vacuum implies that the pressure of the atmosphere is 26 inches of mercury above the pressure in the condenser. (2) Per cent of vacuum, by which is meant the ratio of the observed vacuum to the atmospheric pressure. Thus with the barometer standing at 30 inches a vacuum of 26 inches may be expressed as $100 \times \frac{26}{30} = 86.6$ per cent vacuum. This method of expression gives an idea of the

* See "The Influence of Air on Vacuum in Surface Condensers," *Engng.*, April 17, 1908; *Power*, Feb. 2, 1909, p. 235.

efficiency of the condensing system. For example, the degree of vacuum indicated by 26 inches would be 93 per cent with a barometric pressure of 28 inches but only 84 per cent when the barometer reads 31 inches. (3) Absolute pressure. Thus a 26-inch vacuum referred to a 30-inch barometer would be indicated as a pressure of $30 - 26 = 4$ inches absolute, or 1.9 pounds per square inch.

207. Function of the Condenser. — The function of a condenser in connection with a steam engine or turbine is primarily the reduction of back pressure, though in some instances, notably in marine work, the recovery of the condensed steam may be of equal importance. The advantages to be gained by decreasing back pressure may be most readily illustrated by the following example: A non-condensing engine taking steam at a pressure of 100 pounds absolute and cutting off at one-quarter stroke will have, theoretically, a mean effective pressure on the piston of 44.6 pounds per square inch, the back pressure being 14.7 pounds per square inch absolute. If the engine exhausts into a condenser against a 26-inch vacuum (1.7 pounds absolute) the mean effective pressure will be increased to $44.6 + (14.7 - 1.7) = 57.6$ pounds per square inch, resulting in a gain in power which may be expressed

$$\text{H.P.} = \frac{P_r A S}{33{,}000}, \tag{82}$$

in which

H.P. = horse power gained.

P_r = reduction in back pressure, pounds per square inch.

A = area of the piston in square inches.

S = piston speed in feet per minute.

If P = mean effective pressure on the piston when running non-condensing, the percentage of increase of power may be expressed

$$\text{Per cent} = 100\frac{P_r}{P}. \tag{83}$$

In the above example the percentage of power gained would be

$$100\frac{13}{44.6} = 29.2 \text{ per cent.}$$

The actual gain due to the use of the condenser would be much less than this, depending upon the type of engine and conditions of operation, as shown in the results of engine performances outlined in Chapter X.

TABLE 49.

PRESSURE OF AQUEOUS VAPOR IN INCHES OF MERCURY FOR EACH DEGREE F.

(Marks and Davis.)

	0°	1°	2°	3°	4°	5°	6°	7°	8°	9°
30°.........	......	......	.180	.188	.195	.203	.212	.220	.229	.238
40°.........	.248	.257	.268	.278	.289	.300	.312	.324	.336	.349
50°.........	.362	.376	.390	.405	.420	.436	.452	.468	.486	.503
60°.........	.522	.541	.560	.580	.601	.622	.644	.667	.690	.714
70°.........	.739	.764	.790	.817	.845	.873	.903	.964	.996	1.03
80°.........	1.03	1.06	1.10	1.13	1.17	1.21	1.25	1.30	1.33	1.37
90°.........	1.42	1.46	1.51	1.55	1.60	1.65	1.71	1.76	1.81	1.87
100°.........	1.93	1.98	2.04	2.11	2.17	2.24	2.30	2.37	2.44	2.51
110°.........	2.60	2.66	2.74	2.82	2.90	2.99	3.07	3.16	3.25	3.34
120°.........	3.44	3.53	3.63	3.74	3.84	3.95	4.06	4.17	4.28	4.40
130°.........	4.52	4.64	4.76	4.89	5.02	5.16	5.29	5.43	5.58	5.73
140°.........	5.88	6.03	6.18	6.34	6.51	6.67	6.84	7.02	7.20	7.38

With steam turbines the advantage gained by reduction of back pressure is more marked than with the reciprocating engine, though theoretically the same for the same range of expansion. Initial condensation, leakage past valves, and other sources of loss prevent a reciprocating engine from benefiting from a good vacuum to the same extent as a turbine. (See paragraph 205.)

Referring again to the example given above, if the steam is cut off at about one-sixth stroke, the work done when running condensing will be the same as when running non-condensing and cutting off at one-quarter. Theoretically the steam consumption will be decreased nearly in proportion to the reduction in cut-off. Generally speaking, a condensing engine will require from 20 to 30 per cent less steam for a given power than a non-condensing engine. (See results of engine tests, paragraph 179.) This decrease in steam consumption is only an apparent one. If steam is used by the auxiliaries in creating the vacuum, the amount must be added to that consumed by the engine, unless the steam exhausted by the former is utilized to warm the feed water, in which case only the difference between the heat entering the auxiliaries and that returned to the heater should be charged against the engine. The power necessary to operate the condenser auxiliaries varies from one to six per cent of the main engine power, depending upon the type and conditions of operation. (See paragraph 228.)

In power plants where the exhaust steam is not used for heating or manufacturing purposes, the engines are almost invariably operated condensing, provided there is an abundant supply of cooling water. Even if the water supply is limited, it is often found to be economical to

use some artificial cooling device, notwithstanding the high first cost and cost of operation of the latter.

Some of the considerations affecting the propriety of running condensing and the choice of condensing systems are taken up in paragraphs 230 and 231.

The Law of Condensation of Steam: Pro. Inst. of Civil Engrs., Nov. 30, 1897; Engr., Lond., Dec. 17, 1897, p. 609. *Relation of Pressure and Temperature in Condensers*: Power, April, 1902, p. 28, June, 1902, p. 30. *The Measurement of Vacuum*: Engr., Lond., April 21, 1905. *Experiments on Condensation of Steam*: Engr., Oct. 15, 1897, p. 481. *Condensation Fallacies and Facts*: Machinery, Sept., 1904, p. 38; Elec. Review, June 17, 1904. *Condenser Pressure, a Neglected Point in Steam Condensers*: Mech. Engr., Sept. 30, p. 484.

Advantages of Condensing: Amer. Elecn., Sept., 1904, p. 469. *Condensers for Steam Engines and Turbines* (F. Foster): Mech. Engr., Oct. 28, 1905, pp. 637, 655. *Condensers, to What Extent Should They be Used*: Power, Nov., 1899. *The Value of the Condenser*: Power, Oct., 1902. *The Effects of Vacuum on Steam Engine Economy*: Eng. Mag., June, 1905. *Importance of Condensers*: Power, May, 1897; Evaporating, Condensing, and Cooling Apparatus (E. Hausbrand), published by Scott, Greenwood & Co., London.

The Use of Condensers: Electrician, Lond., Oct. 12, 1900.

208. Classification of Condensers. — The following is a classification of a few well-known steam engine condensers:

1. Jet condensers.	Parallel current (*a*)	Ordinary (1)	Worthington.
			Blake.
			Deane.
		Siphon (2)	Baragwanath.
			Bulkley.
		Ejector (3)	Schutte.
	Counter current (*b*)	Barometric	Weiss.
			Alberger.
			Tomlinson.
2. Surface condensers.	Water cooled (*a*)	Single-flow	Baragwanath.
		Double-flow	Wheeler.
		Multi-flow	Wainwright.
	Air cooled (*b*)	Forced draft	Fouche.
		Natural draft	Pennell.
	Evaporative (*c*)		Ledward.

Condensers may be divided into two general groups:

1. *Jet condensers*, in which the steam and cooling water mingle and the steam is condensed by direct contact, Figs. 203 to 211.

2. *Surface condensers*, in which the steam and cooling medium are in separate chambers and the heat is abstracted from the steam by conduction, Figs. 212 to 215.

Jet condensers may be further grouped into two classes, according to the direction of flow of the air and cooling water:

(*a*) *Parallel-current condensers,* in which the condensed steam, cooling water, and air flow in the same direction, collect at the bottom of the condenser chamber, and are exhausted by a suitable pump, Fig. 203.

(*b*) *Counter-current condensers,* in which the cooling water and condensed steam flow from the bottom of the chamber, usually by gravity, while the air is drawn off at the top, Fig. 229b.

Parallel-current condensers may be subdivided into three classes:

(1) *Standard condensers,* in which the cooling water, condensed steam, and air are exhausted by a vacuum pump, Figs. 203 and 209.

(2) *Siphon condensers,* in which the cooling water, condensed steam, and air are exhausted by a barometric column, Fig. 205.

(3) *Ejector condensers,* in which the condensed steam and air are exhausted by the cooling water, on the ejector principle, Fig. 206.

Surface condensers may be classified according to the nature of the cooling medium as

(*a*) *Water-cooled condensers,* Fig. 212.

(*b*) *Air-cooled condensers,* Fig. 217..

(*c*) *Evaporative condensers,* in which the condensation of the steam is brought about by the evaporation of a fine stream of water trickling on the outside of the tubes.

209. Ordinary Jet Condensers. — Fig. 203 shows a section through a Worthington jet condenser, illustrating the parallel-current principle. When the pump is started a partial vacuum is created in the suction chamber above the valves *H*, *H* in the cone *F*. As soon as sufficient air has been exhausted, cooling water enters at *B* with a velocity depending upon the degree of vacuum in chamber *F* and the suction head, and is divided into a fine spray by the adjustable serrated cone *D*. The spray mingles with the exhaust steam entering at *A* and both move downwards with diverse velocities. The steam gives up its heat to the water and condenses. The velocity of the steam diminishes in its downward path to zero, while the velocity of the water increases according to the laws of falling bodies. The condensed steam, cooling water, and air collect at the lower part of the condenser and are exhausted by

the *wet air pump G*, from which they are forced through opening J to the hot well. The vacuum in chamber F will depend upon the vapor tension of the warm water in the bottom of the well, the amount of air carried along by the cooling water and steam, and the tightness of

Fig. 203. Worthington Independent Jet Condenser.

valves and joints. In case the water accumulates in the condenser cone F, either by reason of an increased supply or by a sluggishness or even stoppage of the pump, the condensing surface is reduced to a minimum, as soon as the level of the water reaches the spray pipe and the spray becomes submerged, and only a small annular surface of water is exposed to the exhaust steam. The vacuum is immediately broken, and the exhaust steam escapes by blowing through the injection pipe

and through the valves of the pump and out the discharge pipe at *J*, forcing the water ahead of it; consequently flooding of the steam cylinder cannot occur. In starting up the condenser a partial vacuum for inducing a flow of injection water into the condenser chamber may be created by the pump if the suction lift is not too great. Many engineers, however, prefer to install a small forced injection or priming pipe the function of which is to condense sufficient steam to produce the necessary partial vacuum. Fig. 222 shows such an installation.

FIG. 204. Section through a Blake Jet Condenser.

Fig. 204 shows a section through the condensing chamber and air pump of a Blake vertical jet condenser with an automatic vacuum-breaking device. The injection water enters at opening marked "injection" and flows through the adjustable "spray" nozzle in a fine spray, at an angle of about 45 degrees, and impinges on the conical sides of the upper condenser chamber. The spray falls from the sides to the projecting ledges shown in the illustration. The ledges prevent the

spray from falling directly to the bottom of the chamber and insure an efficient mingling of steam and cooling water. A perforated copper plate is substituted for the shelves when the force of the injection water is not sufficient to produce spray. The circulating water and condensed steam together with the non-condensable gases are drawn off at the bottom of the chamber. The vacuum-breaking device is shown at the right of the figure. When the rising water reaches the level of the float chamber, as in the case of an accidental stoppage of the air pumps, the float is raised and forces a check valve from its seat and allows an inrush of air to break the vacuum, thus preventing further suction of water into the condenser and consequent flooding of the engine. A is the forced injection or "priming" inlet used in starting up when the suction lift is considerable.

Condenser Types and Applications: Power, June, 1906, p. 44; Engr. U.S., Jan., 1906, pp. 55–66. *Jet Condensers* (McBride): Trans. A.S.M.E., Vol. 12, p. 187; American Machinist, March 7, 1895, p. 185; Engr. U.S., Jan., 1906, p. 61; Whitham's Steam Engine Design, p. 294; Seaton's Manual of the Marine Engine, Chapter XI.

210. Condensing Water, Jet Condensers. — In a jet condenser the cooling water and exhaust steam mingle, and the degree of vacuum is a function of the final or discharge temperature; thus the quantity of cooling water required depends upon its initial temperature, the temperature of the discharge water, and the total heat in the steam entering the condenser. If the steam in the low-pressure cylinder at release is dry and saturated, the heat entering the condenser will correspond to the total heat in steam at release pressure, but it usually contains considerable moisture, part of which is reëvaporated when the exhaust valve opens to the condenser; however, it is sufficiently accurate for all practical purposes to assume the exhaust steam entering the condenser to be dry and saturated and its heat to correspond to the pressure in the condenser.

Let λ = total heat of steam at condenser pressure above 32 degrees.
T_2 = temperature of the discharge water.
T_0 = initial temperature of the cooling water.
W = weight of cooling water in pounds necessary to condense and cool one pound of steam to the required discharge temperature.

Then
$$W = \frac{\lambda - T_2 + 32}{T_2 - T_0}. \tag{84}$$

Example: How many pounds of cooling water are necessary to condense one pound of steam under the following conditions: Barometer 29.92; vacuum 26 inches; temperature of injection water 60 degrees F.

The temperature of aqueous vapor corresponding to an absolute pressure of $29.92 - 26 = 3.92$ inches of mercury is 125 degrees F. (See Table 49.) The discharge temperature, however, must be less than this, as the pressure in the condenser is due not only to the aqueous vapor but to that of the air carried over with the circulating water and the condensed steam. In a condenser of this type the discharge temperature will be from 10 degrees to 15 degrees lower than that corresponding to the vacuum as recorded by the gauge. In this case assume it to be 15 degrees lower, i.e., $T_2 = 125 - 15 = 110$ degrees.

The total heat corresponding to a pressure of 3.92 inches of mercury is 1120 B.T.U. above 32 degrees (see steam tables); $T_0 = 60$ degrees; $T_2 = 110$ degrees.

$$W = \frac{1120 - 110 + 32}{110 - 60} = 20.8.$$

Evidently the higher the temperature of the discharge water the less will be the quantity of cooling water required, and consequently the smaller the weight of air introduced into the condenser; but the warmer the discharge water the greater will be the vapor tension and the lower the degree of vacuum. For reciprocating engines a hot-well temperature between 110 degrees and 130 degrees F. is average practice; with turbines the temperature ranges between 80 degrees and 100 degrees F. On account of the inefficient heat absorption in practical installations, from 5 per cent to 15 per cent is added to the theoretical weight of cooling water as determined from equation (84). Table 50 has been calculated from equation (84).

Cooling Water for Condensers: Am. Mach., May 18, 1905, p. 656; Evaporation and Condensing Apparatus, Hausbrand, pp. 227, 240, 301, 318; Steam Power Plants, Meyer, p. 106.

Wet-Air Pump, Jet Condensers. (See paragraph 285.)

Circulating Pumps. (See paragraph 297.)

211. Effect of Aqueous Vapor upon the Degree of Vacuum. — The futility of attempting to better the vacuum by exhausting the vapor is best illustrated by a specific problem.

Required the volume of aqueous vapor to be withdrawn per hour from a condenser operating under the following conditions, in order that the vacuum may be increased one pound per square inch: Temperature of discharge water 125 degrees; corresponding vapor tension 4 inches of mercury; barometer 30 inches; relative vacuum 26 inches; horse power, 100; steam consumption 20 pounds per horse-power hour; cooling water 25 pounds per pound of steam condensed.

$100 \times 20 \times 25 = 50{,}000$ pounds of cooling water per hour.
$= 833$ pounds of cooling water per minute.

TABLE 50.

RATIO, BY WEIGHT, OF COOLING WATER TO STEAM CONDENSED (THEORETICAL).

(Barometer 29.92.)

Temp. of Injection.	Vacuum 24". Temperature of Steam 141°. Temperature of Hot Well.				
	110	115	120	125	130
40	15.0	13.9	12.9	12.1	11.4
50	17.5	16.0	14.8	13.7	12.8
60	21.0	18.9	17.3	15.8	14.6
70	26.2	23.2	20.7	18.7	17.1
80	35.0	29.8	25.9	23.0	20.5
90	52.4	49.7	34.6	29.5	25.6

Temp. of Injection.	Vacuum 25". Temperature of Steam 134°. Temperature of Hot Well.				
	105	110	115	120	125
40	16.1	14.9	13.8	12.9	12.1
50	19.0	17.4	16.0	14.8	13.7
60	23.2	20.9	18.9	17.2	15.8
70	30.0	26.1	23.0	20.7	18.7
80	42.0	34.8	29.6	25.9	22 8
90	70.0	52.1	41.5	34.5	29.4

Temp. of Injection.	Vacuum 26". Temperature of Steam 125°. Temperature of Hot Well.				
	100	105	110	115	
40	17.5	16.1	14.8	13.8	......
50	21.0	19.0	17.4	16.0	......
60	26.3	23.2	20.9	18.8	......
70	35.0	30.0	26.0	23.0	......
80	57.6	42.0	34.7	29.6	......

Temp. of Injection.	Vacuum 27". Temperature of Steam 114°. Temperature of Hot Well.				
	90	95	100	105	
40	21.2	19.1	17.4	16.0	......
50	26.5	23.4	20.9	19.0	......
60	35.3	30.1	26.2	23.2	......
70	52.9	42.1	34.9	29.8	......
80	......	......	52.3	41.5	......

Temp. of Injection.	Vacuum 27.5". Temperature of Steam 108°. Temperature of Hot Well.				
	80	85	90	95	
40	26.6	23.6	21.1	19.1	......
50	35.6	30.3	26.4	23.4	......
60	52.3	42.5	35.2	30.0	......
70	......	70.8	52.8	42.0	......

Temp. of Injection.	Vacuum 28". Temperature of Steam 100°. Temperature of Hot Well.				
	75	80	85	90	
40	30.5	26.6	23.5	21.1	......
50	42.7	35.5	30.2	26.3	......
60	71.2	53.2	42.3	35.1	......
70	......	......	70.6	52.7	......

Temp. of Injection.	Vacuum 28.5". Temperature of Steam 90°. Temperature of Hot Well.				
	60	65	70	75	
35	42.2	35.8	30.6	29.2	......
40	54.0	43.0	35.6	33.4	......
45	72.0	53.5	42.8	38.8	......
50	......	72.0	53.5	46.6	......

Temp. of Injection.	Vacuum 29". Temperature of Steam 77°. Temperature of Hot Well.				
	55	60	65	67	
35	52.0	43.0	35.8	33.4	......
40	69.3	54.0	43.0	38.4	......
45	......	71.5	54.0	47.0	......
50	......	......	72.0	61.0	......

Now to increase the vacuum one pound per square inch, approximately 2 inches of mercury, the temperature of the water must be lowered to 102 degrees F., that is, 833 (125 − 102) = 19,159 B.T.U. must be abstracted from the water in one minute, or $\frac{19,159}{1030} = 18.6$ pounds of water to be evaporated per minute. (1030 = average heat of vaporization of water under 26 to 28 inches of vacuum.) Now, one pound of vapor at 102 to 125 degrees F. has an average volume of 270 cubic feet.

Therefore 18.6 × 270 = 5022 cubic feet of vapor must be exhausted per minute to increase the vacuum from 26 to 28 inches, which is manifestly impracticable.

212. Injection Orifice. — The velocity of water entering a jet condenser, neglecting friction, may be determined from the formula

$$V = \sqrt{2\,gh}, \tag{85}$$

where

V = velocity of the water in feet per second.
g = acceleration of gravity = 32.2.
h = total head in feet.

If p = pressure below the atmosphere in pounds per square inch,
h_1 = distance in feet between the source of supply and the injection orifice,

then

$$h = 2.3\,p \pm h_1, \tag{86}$$

and equation (85) may be written

$$V = 8.025\sqrt{2.3\,p \pm h_1}. \tag{87}$$

If the supply is under pressure, h_1 is positive; if under suction, it is negative.

Example: What is the theoretical velocity of water entering a condenser with 26-inch vacuum (referred to 30-inch barometer); suction head 8 feet

Here p = pressure in pounds per square inch, corresponding to 26 inches of mercury = 12.8 pounds per square inch.

$h_1 = 8.$
$V = 8.025\sqrt{2.3 \times 12.8 - 8}$
$= 37.1$ feet per second
$= 2226$ feet per minute.

In proportioning the injection orifice in practice the maximum velocity of flow is assumed to be between 1500 and 1800 feet per minute,

or, approximately, area of injection orifice in square inches = weight of injection water in pounds ÷ 650 to 780. ("Manual of Marine Engineering," Seaton, p. 204.) A rough rule gives area of orifice = area of low-pressure piston in square inches ÷ 250. (Seaton, p. 204.)

213. Volume of the Condenser Chamber. — According to Thurston the volume of a jet condenser should be from one-fourth to one-half that of the low-pressure engine cylinder. ("Steam Engine Manual," Thurston, II, 127.)

According to Hutton the volume should not be less than that of the air pump and should approximate three-fourths of that of the engine cylinder in communication with it.

214. Injection and Discharge Pipes. — In practice the diameter of the injection pipe is based on a velocity of 400 to 600 feet per minute and that of the discharge pipe of 200 to 400 feet per minute; the lower figures for pipes under 8 inches in diameter, the upper range for larger diameters.

(Atmospheric relief valves. — See paragraph 351.)

215. Siphon Condensers. — Fig. 205 shows a section through a Baragwanath siphon condenser, illustrating the principles of a parallel-current barometric condenser. The cooling water enters the side of the condenser chamber at *A* and passes downward in a thin annular sheet around the hollow cone *D*. The exhaust steam enters at *B* and is given a downward direction by the goose neck *C*. It flows through the nozzle *D* and is condensed within the hollow cone of moving water, the combined mass including the entrained air discharging through the contracted throat *E* at high velocity into the tail pipe *F*. The water column in the tail pipe must be enough to overcome the pressure of the atmosphere; i.e., it should be 34 feet or more above the surface of the hot well, otherwise water would rise within this pipe to a height corresponding to that of the barometer, which is approximately 34 feet for a barometric pressure of 30 inches of mercury. This is not strictly true when the condenser is in full operation, as the injector effect of the

Fig. 205. Baragwanath Siphon Condenser.

moving mass is sufficient to overcome several pounds pressure, and the tail pipe may be less than 34 feet, but to provide against any possibility of the water being drawn into the cylinder of the engine the length is made greater than 34 feet. The spray cone D is adjustable and admits of close regulation of the water supply without changing the annular form of the stream. The condensing water may be supplied under pressure or under suction. For lifts not greater than 15 feet no supply pump is necessary, the water being raised by the siphon action of the condenser. This condenser requires the same amount of cooling water per pound of steam as the standard jet condenser, and is capable of maintaining a vacuum of from 24 to 25 inches. A vacuum of 28½ inches has been recorded for a condenser of this general type. (Trans. A.S.M.E., 26–388.) An atmospheric relief valve G is provided in case the vacuum fails from any cause, which will permit the steam to escape to the atmosphere.

The above type of condenser is adapted to very muddy cooling water, since no filtration is necessary beyond the removal of such solid matter as may clog up the annular space H.

In the Armour Glue Works at Chicago condensers of this type are successfully maintaining a 90 per cent vacuum with cooling water at 60 degrees F., and the circulating water is practically liquid mud.

Siphon Condensers, Discussion: Trans. A.S.M.E., Vol. 26, p. 388. *Siphon Condensers:* Electrical World, June, 1897, p. 818; Hutton, The Mechanical Engineering of Power Plants, p. 106; Engr. U.S., Jan., 1906.

216. Size of Siphon Condensers. — The size of siphon is indicated by the diameter of the engine exhaust pipe.

Table 51 gives the sizes of barometric condensers as manufactured by prominent makers.

TABLE 51.

SIZE OF SIPHON CONDENSERS.

Steam to be Condensed.		Size Usually Furnished, Inches.	Steam to be Condensed.		Size Usually Furnished, Inches.
Pounds per Hour.	Pounds per Minute.		Pounds per Hour.	Pounds per Minute.	
2,000	33	5	8,000	133	10
3,000	50	7	10,000	166	12
4,000	66	8	15,000	250	14
5,000	83	9	20,000	333	14
6,000	100	9			

Vacuum 26 inches; barometer 30 inches.

The diameter of the throat may be closely approximated by the empirical formula

$$\text{Diam. in inches} = 0.0077\sqrt{Ww}, \tag{88}$$

in which

W = weight of steam to be condensed per hour.
w = weight of water required to condense one pound of steam.

The maximum width of the annular opening for the admission of water may be obtained from the empirical formula

$$\text{Width in inches} = \frac{Ww}{39{,}550\,d}, \tag{89}$$

in which

d = diameter of the nozzle or bottom of the cone in inches.
W and w as in (88)

217. Ejector Condenser. — Fig. 206 shows a section through a Schutte exhaust steam "induction" condenser, illustrating the principles of the ejector condenser in which the

FIG. 206. Schutte Ejector Condenser.

FIG. 207. Piping for Schutte Ejector Condenser.

momentum of flowing water ejects the discharge without the aid of the circulating pump. Exhaust steam enters the ejector through the opening marked "exhaust," passes through a series of inclined orifices and

nozzles at considerable velocity, and, meeting the cooling water in the inner annular chamber, is condensed. The cooling water is drawn in continuously through the opening marked "water," by virtue of the vacuum formed, and sufficient velocity is imparted to the jet to discharge the combined mass of condensed steam, cooling water, and air against the pressure of the atmosphere.

Adjustment for capacity is effected by raising or lowering the ram *R* by means of the wheel *H*. An adjustable sleeve controls the available area of the exhaust inlet by covering more or less openings in the combining tube. When the cooling water is supplied under pressure the openings marked "steam" and *O* are blanked. When water is taken under suction and water under pressure is available for starting, *O* is blanked and opening marked "steam" is connected with the pressure supply. When water is taken under high suction and live steam is used for starting, inlet marked "steam" is connected to live steam and an overflow check valve is placed at *O*. Fig. 207 gives an outline of the necessary piping for a condenser installation of this type. These condensers are made in all sizes conforming with exhaust pipe diameters of 1½ to 20 inches. The same amount of cooling water is required as for jet condensing and vacua of 20 to 25 inches are readily obtained.

Exhaust Steam Induction Condensers: Power, Dec., 1898, p. 14. *Ejector Condenser:* Hutton, Mechanical Engineering of Power Plants, p. 111; Eng. News, Oct. 5, 1905, p. 360.

218. Barometric Condensers.*—Fig. 208 shows a section through a Weiss counter-current condenser, illustrating the principles of a barometric jet condenser. The cooling water enters the upper part of the condensing chamber *A* through pipe *N* and falls in cascades, as shown in the figure, to tail pipe *B*, from which it flows by gravity to the hot well. The exhaust steam enters chamber *A* through pipe *D*,

FIG. 208. Weiss Counter-Current Condenser.

* The author has been informed that the word "Barometric" in connection with jet condensers is the registered trade mark of the Alberger Condenser Company.

and, coming in contact with the cold-water spray, is condensed. The air is exhausted from the top of the condenser by a dry vacuum pump through pipe *F*. In flowing to the pump the air passes upwards through the water spray and its temperature is lowered to that of the injection water, thereby reducing the volume to be exhausted. Any moisture passing over with the air is separated at *G* before reaching the air pump, and flows out through the small barometric tube *H*. The cooling water is forced to the condenser chamber through pipe *N* by any positive displacement pump, the actual head pumped against

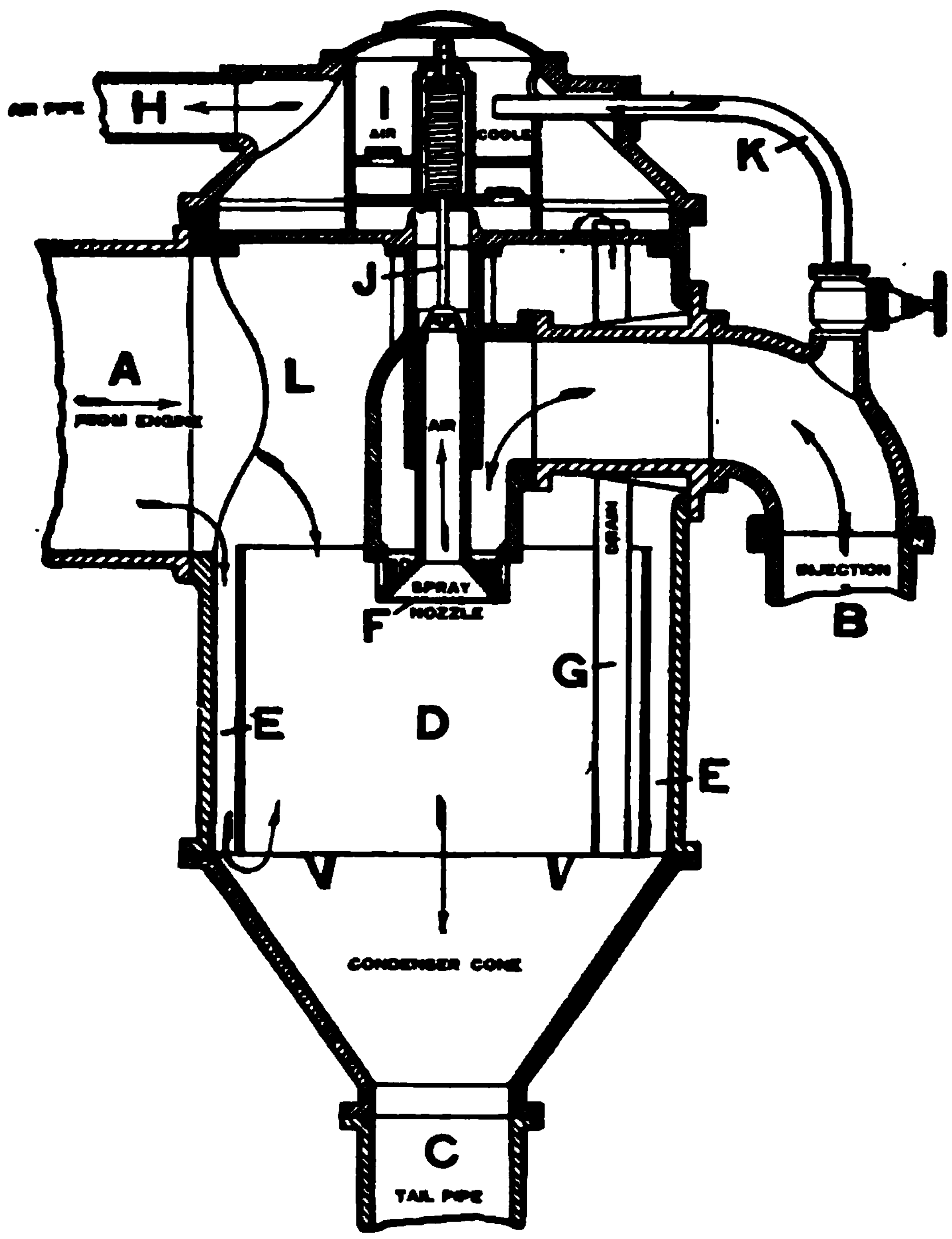


Fig. 209. Section Through Condensing Chamber, Alberger Barometric Condenser.

being the difference between the total height and that of a column of water corresponding to the degree of vacuum in the condenser. The main barometric tube or tail pipe *B* through which the water is discharged is 34 feet or more in length and is provided with a foot valve *C*. The counter-current principle permits a much higher temperature of hot well for the same degree of vacuum than does the parallel current, a hot-well temperature of 120 degrees and a vacuum of 27 inches being readily maintained. A small pipe *K* connecting the main condenser

with the small barometric tube H insures at all times a sufficient quantity of water in the small auxiliary hot well to seal the tube. The water from this auxiliary hot well flows over a weir, as indicated, into a counter-weighted bucket M, the latter having a hole in the bottom which allows the normal flow to escape. But in case a sudden heavy overload is thrown on the engines, and the adjustment is for a light load, the temperature of the discharge will reach the boiling point and an abnormal quantity of water will flow down the small barometric tube. This will cause the water to flow into the bucket much faster than the opening in the bottom can dispose of it; as a result the bucket will increase in weight and will open up a free-air valve L which reduces the vacuum two or three inches and raises the boiling point without " dropping " the vacuum entirely. E is the atmospheric relief valve.

Fig. 209 shows a section through the condensing chamber of an Alberger barometric condenser. In principles of operation the condenser is similar to the Weiss, but differs considerably in details. Exhaust steam enters at A and divides into two streams, one flowing directly to the inner chamber D, the other through the annular space E. Cooling water enters through B and is broken up into a fine spray by the serrated cone F, which is hung upon a long spring, thus automatically adjusting itself to the quantity of water entering the condenser. After condensing the exhaust steam in the inner cylinder the partly heated spray of cooling water in falling is brought in contact with the exhaust steam which enters through the annular space. This process permits of a high hot-well temperature without affecting the degree of vacuum. The air which is not entrained by the cooling water and carried down the tail pipe collects under the spray cone F and ascends through the tubular support of the cone into the air cooler. This air cooler is simply a small chamber in which the non-condensable gases are cooled by a small portion of the circulating water before they are withdrawn by the air pump. The circulating water used for the purpose is forced into the cooling chamber through pipe K and falls through serrated openings in the bottom to the condenser proper. The air enters the chamber through these same openings, and is withdrawn by the air pump. Surrounding the cooler is a separating space of large capacity to allow the subsidence of any entrained moisture before the air reaches the vacuum pump.

Fig. 236 shows a typical installation of an Alberger condenser in connection with a cooling tower, and Fig. 226 that of a Weiss condenser in the Northwestern Elevated R. R. Power Station, Chicago.

Fig. 210 shows a section through the condensing chamber of a Worthington barometric condenser. The drawing is self-explicit.

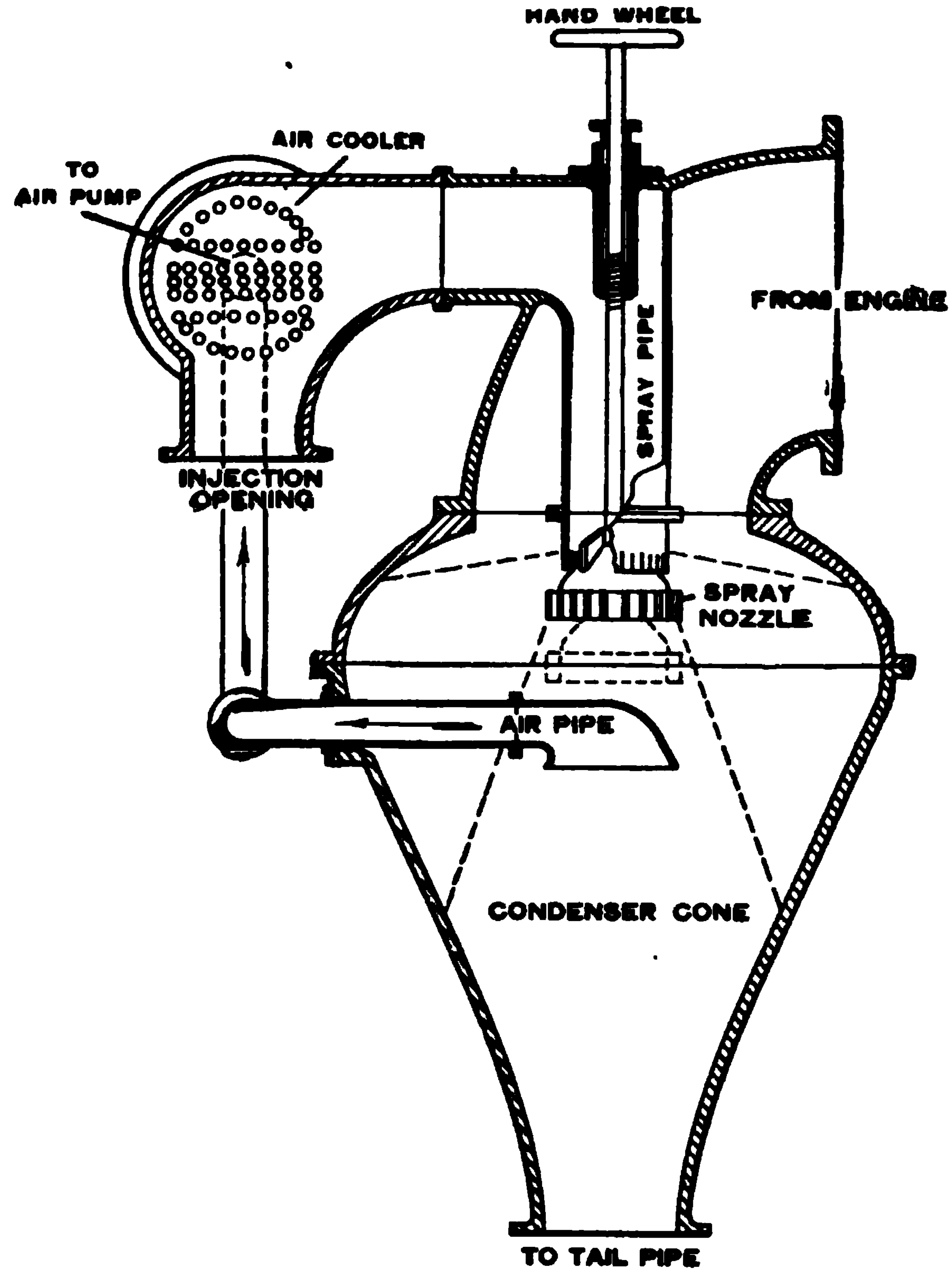


FIG. 210. Section Through Condensing Chamber, Worthington Barometric Condenser.

Fig. 211 shows a section through a Tomlinson barometric condenser. The air pump instead of discharging into the atmosphere is made to deliver into the tail pipe where the vacuum is still sufficient to support the column of water below the point of delivery. The effect produced is that of a two-stage air pump, the tail pipe becoming the second stage. Suitable by-pass valves enable the air pump to be discharged into the atmosphere or to be cut out entirely. (*Power*, February, 1907, p. 94.)

Fig. 211a shows the application of a centrifugal pump to the tail pipe of a barometric condenser. This permits of a very short tail pipe, as the pump takes the place of the barometric column.

Counter-Current Condensers : Am. Elecn., Feb., 1905, p. 81; Power, March, 1905, p. 182, Jan., 1906, p. 44; Engr. U.S., Jan., 1906, p. 58; Hausbrand, Evaporating and Condensing Apparatus, Chapter XX; Bulletin No. 6, Heisler Mfg. Co., St. Marys, Ohio. *The Barometric Condenser :* Power, Jan., 1907, p. 1.

FIG. 211. Tomlinson Type B Barometric Condenser.

As previously outlined, surface condensers may be divided into three general classes, (*a*) water cooled, (*b*) air cooled, and (*c*) evaporative.

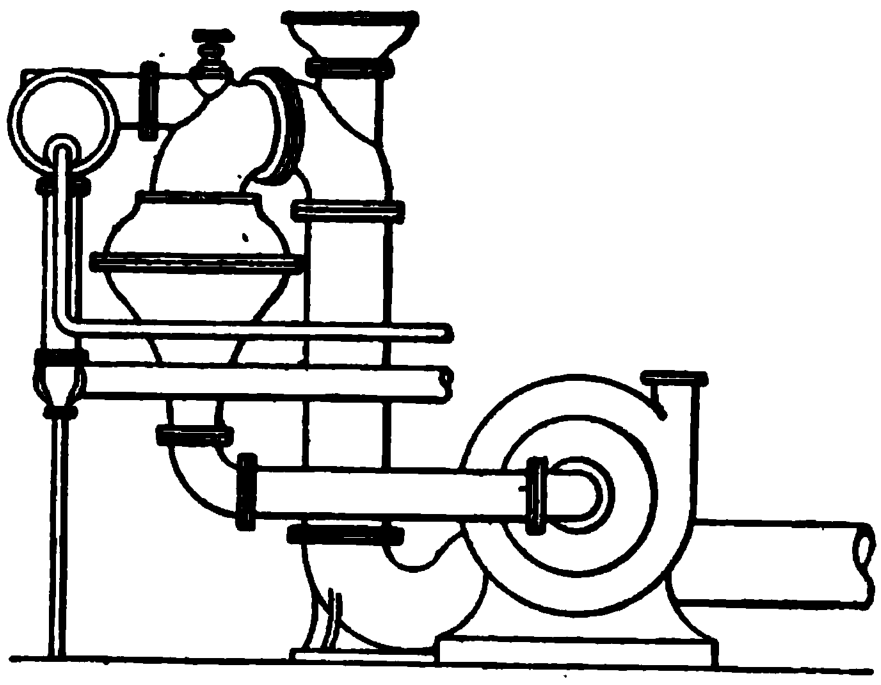


FIG. 211a. Centrifugal Pump Applied to the Tail Pipe of a Barometric Condenser.

219. Water-Cooled Surface Condensers. — Water-cooled surface condensers are by far the most extensive in use and only occasionally are the conditions such as to warrant the installation of the other class. They are ordinarily classified as (1) single-flow, (2) double-flow, and (3) multi-flow.

Fig. 212 shows a sectional elevation through a Baragwanath vertical condenser, illustrating the single-flow type. It consists essentially of a cast-iron shell provided with two heads, into which a number of one-inch brass tubes are expanded. Exhaust steam fills the shell and flows around and between the tubes, while the cooling water is caused to circulate through the tubes by means of a circulating pump. The steam is condensed by contact with the tubes and drops to the bottom tube sheet, from which it is exhausted by the air pump. The circulating water flows through the tubes in one direction only, hence the name "single flow." To allow for the unequal expansion of shell and tubes

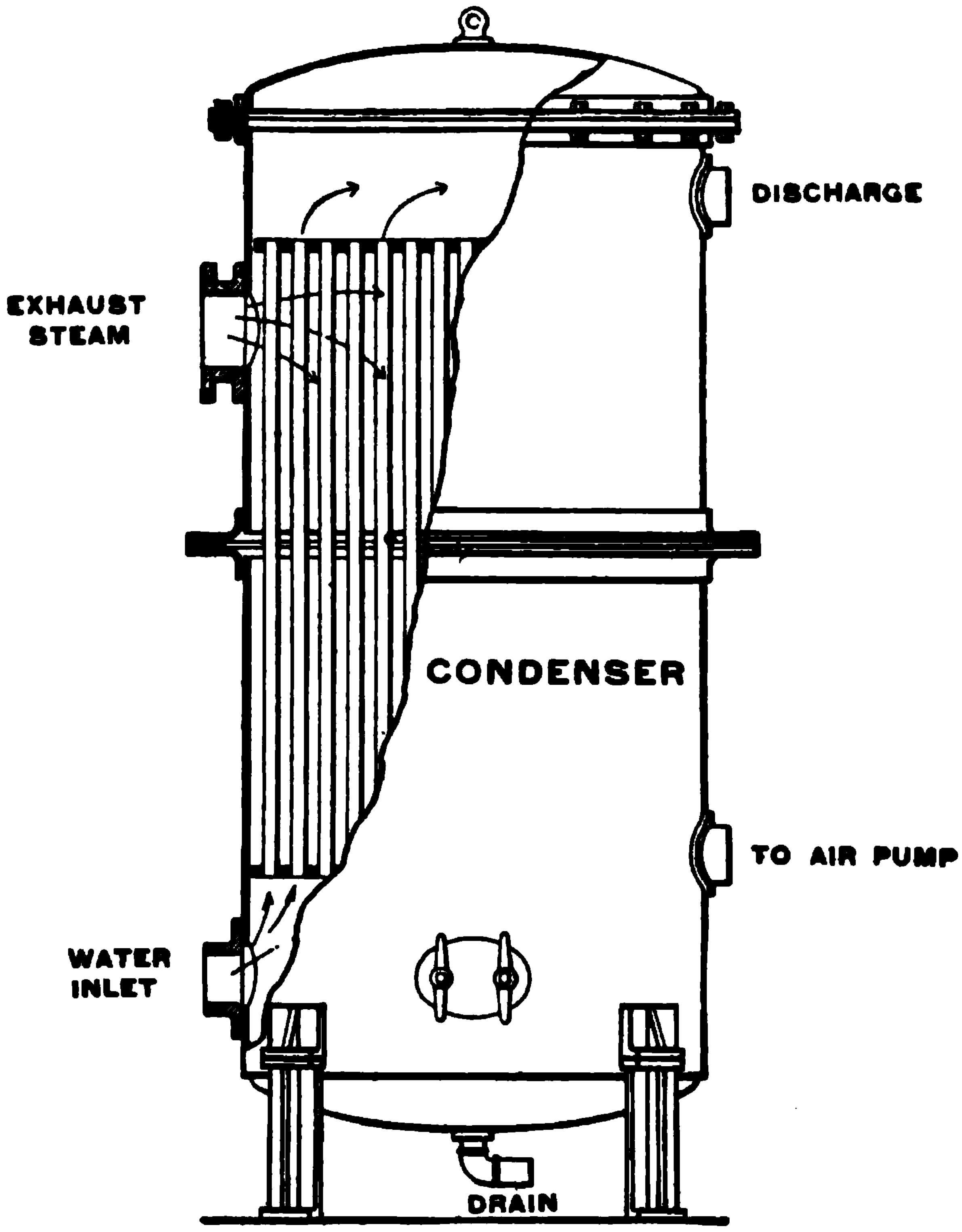


FIG. 212. Baragwanath Surface Condenser.

FIG. 213. Wheeler Surface Condenser and Pumps.

the two halves of the shell are provided with slightly thinner plates flanged outward, the flanges being bolted together with a spacing ring between them. This joint gives to the shell, in the direction of its length, a certain amount of elasticity which is sufficient to allow for the greatest possible elongation of the tubes without straining the tube ends and causing leakage.

Fig. 213 shows a section through a Wheeler admiralty surface condenser mounted on a combined air and circulating pump, illustrating the typical "double-flow" surface condenser. The condenser proper consists of a ribbed cast-iron chamber of rectangular section fitted with a number of small seamless drawn brass tubes through which the cooling water is forced by suitable means. The exhaust steam enters at the top and is prevented from impinging directly against the tubes by baffle plates, which serve also to distribute the steam more evenly over the cooling surface. The steam in passing between the tubes is condensed, and falls to the bottom of the chamber, from which it is removed, together with the entrained air, by a vacuum pump. The water chamber between the tube sheet and the head is divided into two compartments, as shown in the illustration, the partition being so arranged that the water flows first through the lower set of tubes and then through the upper set in the opposite direction. Thus the temperature of the cooling water increases as it rises, and reaches a maximum where the exhaust steam enters. Condensation begins as soon as the vapor enters the condenser, and the surfaces of the tubes are at once covered with a thin film of water flowing downwards from tube to tube.

FIG. 214. Surface Condenser, C. H. Wheeler & Co.

Fig. 214 gives the details of a C. H. Wheeler & Co.'s high-vacuum surface condenser. The condensing chamber is of the series-parallel type in which the water enters the top group of tubes, then passes to the middle section and finally through the bottom section.

Connecting chambers are provided at the ends of the shell as illustrated. This construction of water chamber keeps the condenser completely filled with cooling water at all times. The inlet is at the bottom but the water is carried up through the annular chamber to the top of the tubes.

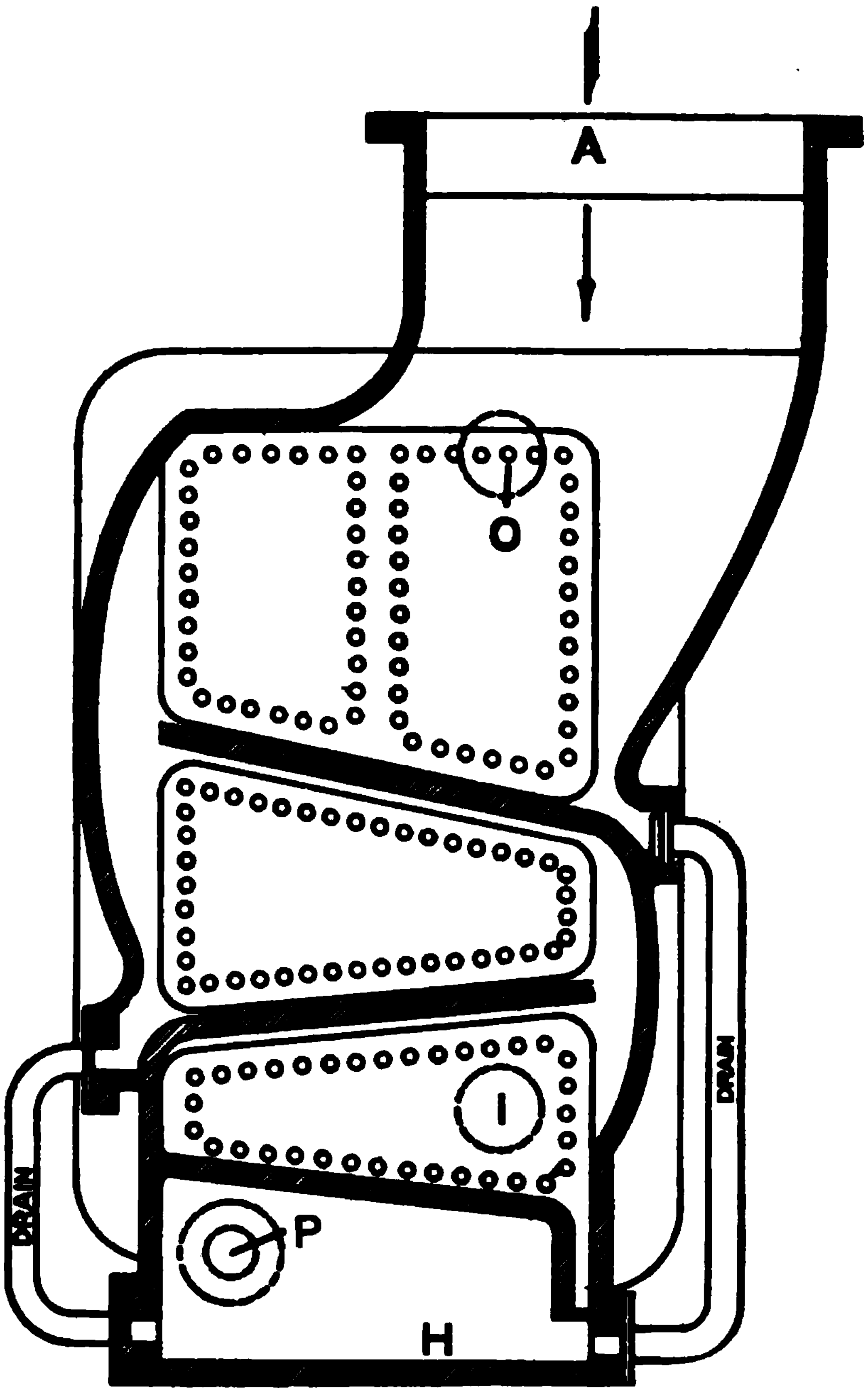


Fig. 215. Weighton Multi-Flow Surface Condenser.

Fig. 215 shows a section through a multi-flow surface condenser designed by Professor R. L. Weighton. The condenser has three compartments separated by two diaphragms inclined to the horizontal. Each compartment is fitted with a number of brass tubes three-fourths inch in diameter by four feet in length, spaced one and one-eighth inches between centers. The cooling water circulates through the tubes five times, giving an effective length of 20 feet. The notable features of the condenser are abolishment of steam space, and compartment drainage of condensed steam. Mere passages of such shape

and section as will insure distribution of the steam over the entire surface are used instead of the large steam space usually associated with surface condensers. Each compartment is ***separately drained*** to the air pump, so that the surfaces in the lower compartments are unimpeded in their condensing action by the condensed steam from the upper compartments flowing over them. Referring to Fig. 215: Exhaust steam enters the condenser at A and flows toward the hot well H. The greater part of the steam is condensed in the first section of the condenser, and the condensation is drained directly to the hot well. The balance of the condensation takes place in the remaining sections, the condensed steam being withdrawn from each section. The wet-air pump withdraws the condensed steam and non-condensable gases through opening P. Cooling water enters at I and leaves at O. An exhaustive series of tests on a condenser of this type credit it with a much higher efficiency than the ordinary single or double-flow apparatus. (See "The Efficiency of Surface Condensers," Proc. Institute of Naval Architects, March, 1906; also *Engineer*, London, April 27, 1906.)

220. Cooling Water, Surface Condensers. — The amount of cooling water required per pound of steam in a surface condenser is dependent upon the vacuum, the temperature of the condensed steam, and the range in temperature of the cooling water; it may be closely approximated from the formula

$$W = \frac{\lambda - T_1 + 32}{T_2 - T_0}, \qquad (90)$$

where

λ = total heat of the exhaust steam above 32 degrees F.
T_1 = temperature of the condensed steam.
T_0 = temperature of the injection water.
T_2 = temperature of the discharge water.
W = pounds of injection water necessary to condense one pound of steam.

Example: Required the quantity of cooling water necessary to condense one pound of steam under the following conditions: Initial temperature of the cooling water 60 degrees F.; final temperature 100 degrees F.; vacuum 26 inches, referred to 30-inch barometer. Here λ = 1120 B.T.U., T_0 = 60, T_2 = 100.

$$W = \frac{1120 - 110 + 32}{100 - 60} = 26.0.$$

That is, the ratio of cooling water to condensed steam is 26.0 to 1. In turbine practice where vacua as high as one-half pound absolute are

obtained, the ratio of cooling water to condensed steam is nearly twice this quantity. For example, if a vacuum of 28.92 inches is desired with the barometer at 29.92 and the range of the circulating water temperature is 70 to 50 degrees and the temperature of the hot well 80 degrees, the ratio will be

$$W = \frac{1106 - 80 + 32}{70 - 50} = 52.9.$$

In determining the amount of cooling water it is well to bear in mind that in the ordinary condenser of the single or double-flow type

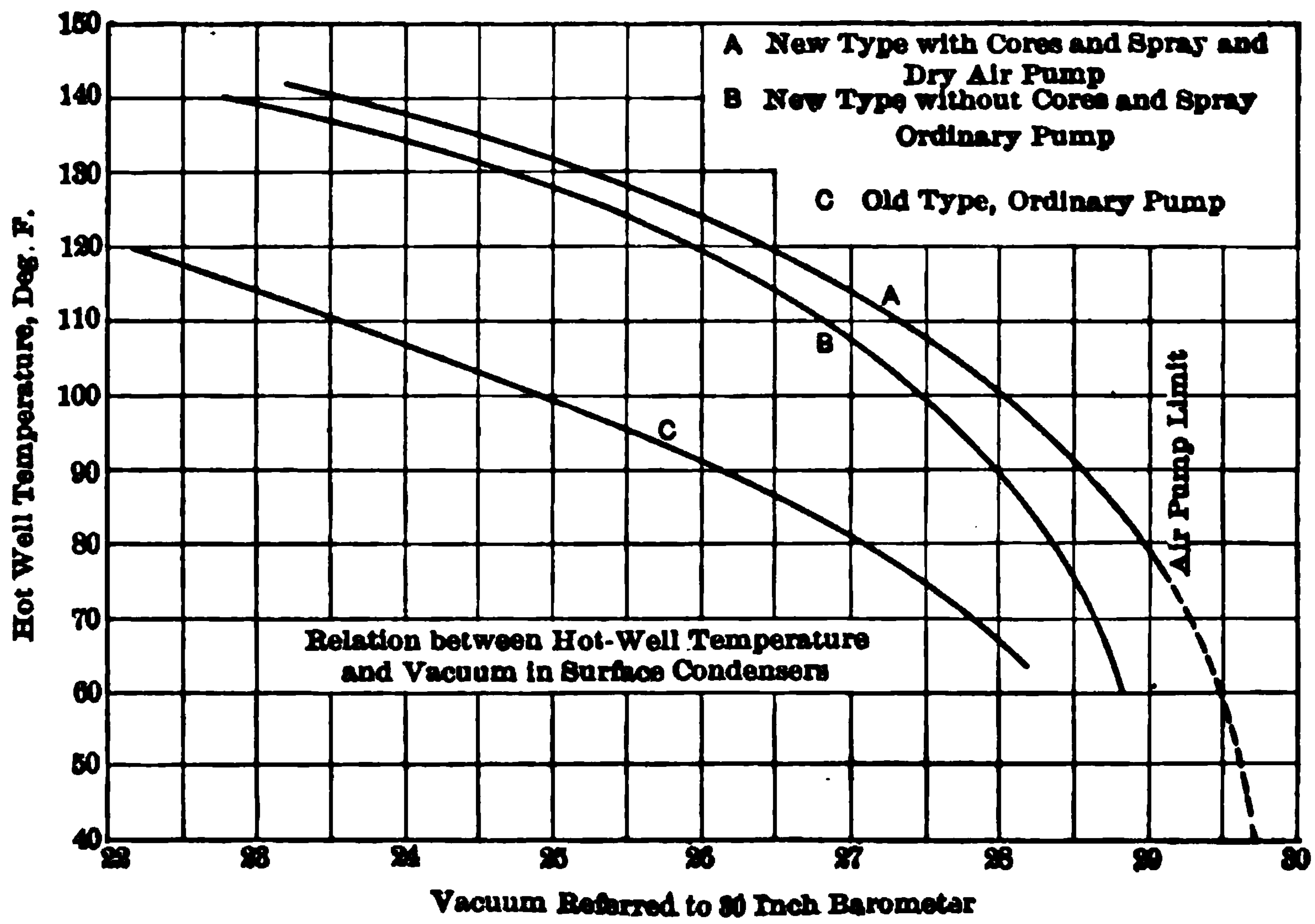


FIG. 216.

the temperature of the condensed steam will be from 10 to 20 degrees lower than that corresponding to the degree of vacuum in the condenser, and that the temperature of the condensing water at the discharge point will be from 5 to 10 degrees lower than the temperature due to the vacuum.

With well-designed condensers of the multi-flow type the temperature of the hot well may be from 3 to 5 degrees higher than the temperature due to the vacuum, and the temperature of the condensing water at the discharge point may be equal to or slightly higher than that due to the vacuum. (Proc. Inst. of Naval Arch., March, 1906.) (See Fig. 216.)

221. Extent of Water-Cooling Surface. — Theoretically the operation of a surface condenser is divided into two periods, (1) the period

of condensation during which the heat of vaporization at the observed pressure is removed and (2) the period of cooling during which the temperature of the condensed steam is reduced. In order to determine accurately the extent of cooling surface it would be necessary to calculate the heat transmission for each of the two periods. In practice, however, it is assumed that condensation and cooling take place simultaneously, and that the mean temperature difference is a direct function of the temperature corresponding to the exhaust steam in the condenser and that of the condensed steam and cooling water. The error in these assumptions has only a slight influence on the estimation of the cooling surface and is entirely lost sight of in the liberal factor allowed in practice.

Let S = cooling surface in square feet.

λ = total heat above 32 degrees of the exhaust steam at condenser pressure.

T_0 = initial temperature of the circulating water.

T_2 = final temperature of the circulating water.

T_1 = final temperature of the condensed steam.

T_s = temperature of the exhaust steam at condenser pressure.

U = coefficient of heat transmission, B.T.U. per hour, per degree difference in temperature, per square foot of cooling surface.

d = mean difference in temperature between T_s and T_2, and T_0.

W = weight of condensed steam per hour.

$$d = \frac{T_2 - T_0}{\log_e \dfrac{T_s - T_0}{T_s - T_2}} \quad \text{(see equation (118), Chap. XII);}$$

and since the heat absorbed by the cooling water is equal to the heat given up by the steam,

$$SUd = W\{\lambda - (T_1 - 32)\}. \tag{91}$$

$$S = \frac{W(\lambda - T_1 + 32)}{Ud}. \tag{92}$$

Whitham ("Steam-Engine Design," p. 283) uses the arithmetic mean $d = T_s - \frac{T_0 + T_2}{2}$ instead of the mean as determined from (118).

Equation 118 is based on the assumption that the fluid on each side of the tube is homogeneous, which is far from being true in the case of the air-steam mixture in a condenser, and for this reason many designers prefer to use the simpler arithmetic formula.

The coefficient of heat transfer, U, as used in above equations refers

to the *mean* or average values for the *entire* surface since the *actual* heat transmission varies widely for different parts of the condenser; thus the actual value of U varies from over 1000 in the first few rows of the tubes (where the steam comes directly into contact with the cooling surface) to less than 50 in the bottom row (where the tubes are practi-

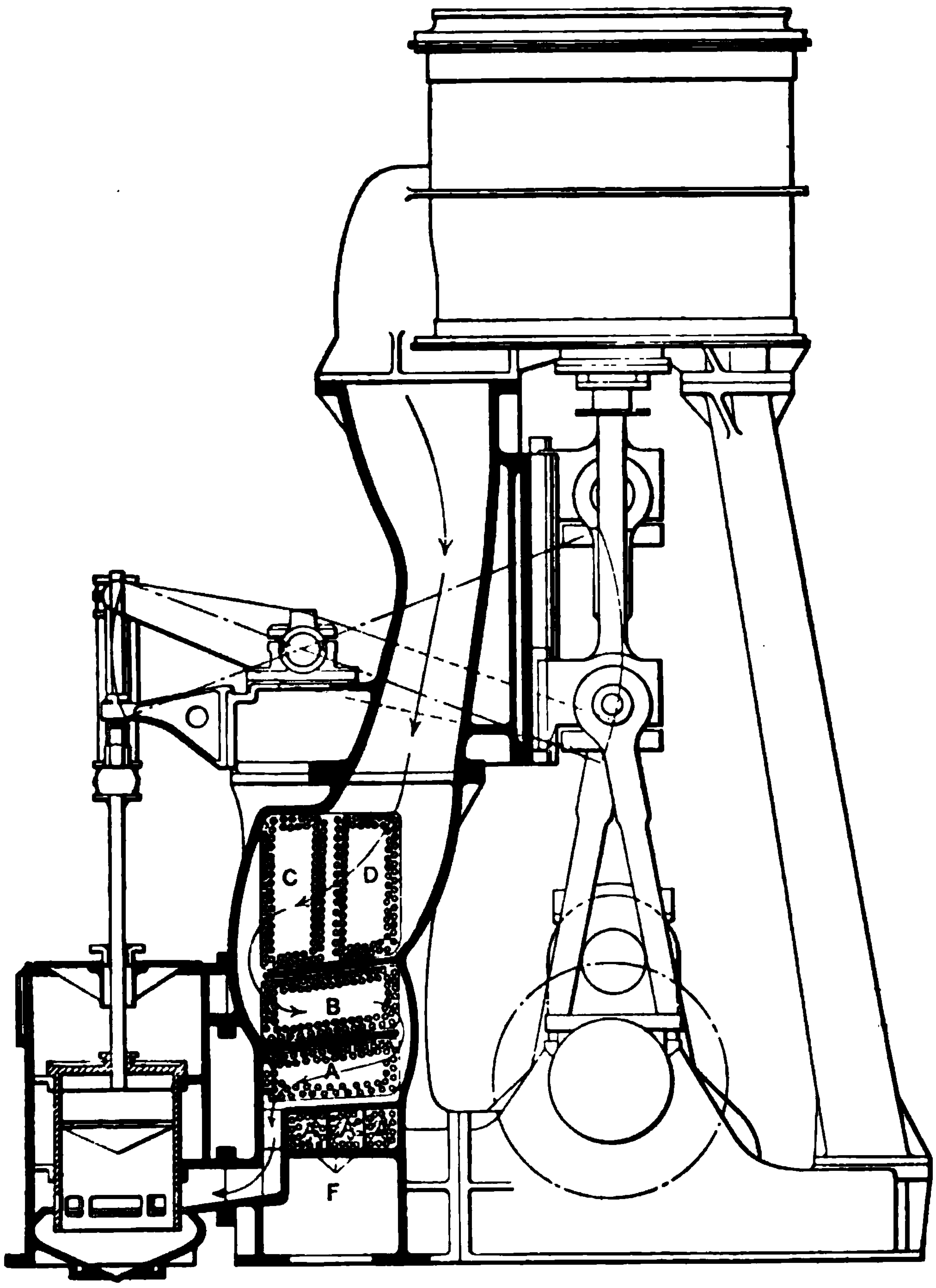


FIG. 216a. Application of Weighton Dry-tube Surface Condenser to Vertical Marine Engine.

cally submerged in water of condensation) and to 3 or less for the tubes surrounded only by air.

Prof. Josse of the Royal Technical School, Charlottenburg, after an exhaustive investigation of the subject found that the *actual* value of U varied with

(1) The material, thickness, shape and cleanliness of the tubes.

(2) The velocity of the water through the tubes.

(3) The velocity of the steam against the tubes.

(4) The percentage of air in the steam surrounding the tubes.

(5) The extent of submersion of the steam side of the tubes.

Some of the results of his investigations are shown in Figs. 216, b, c, and d. See also *Power and Engr.*, Feb. 2, 1909.

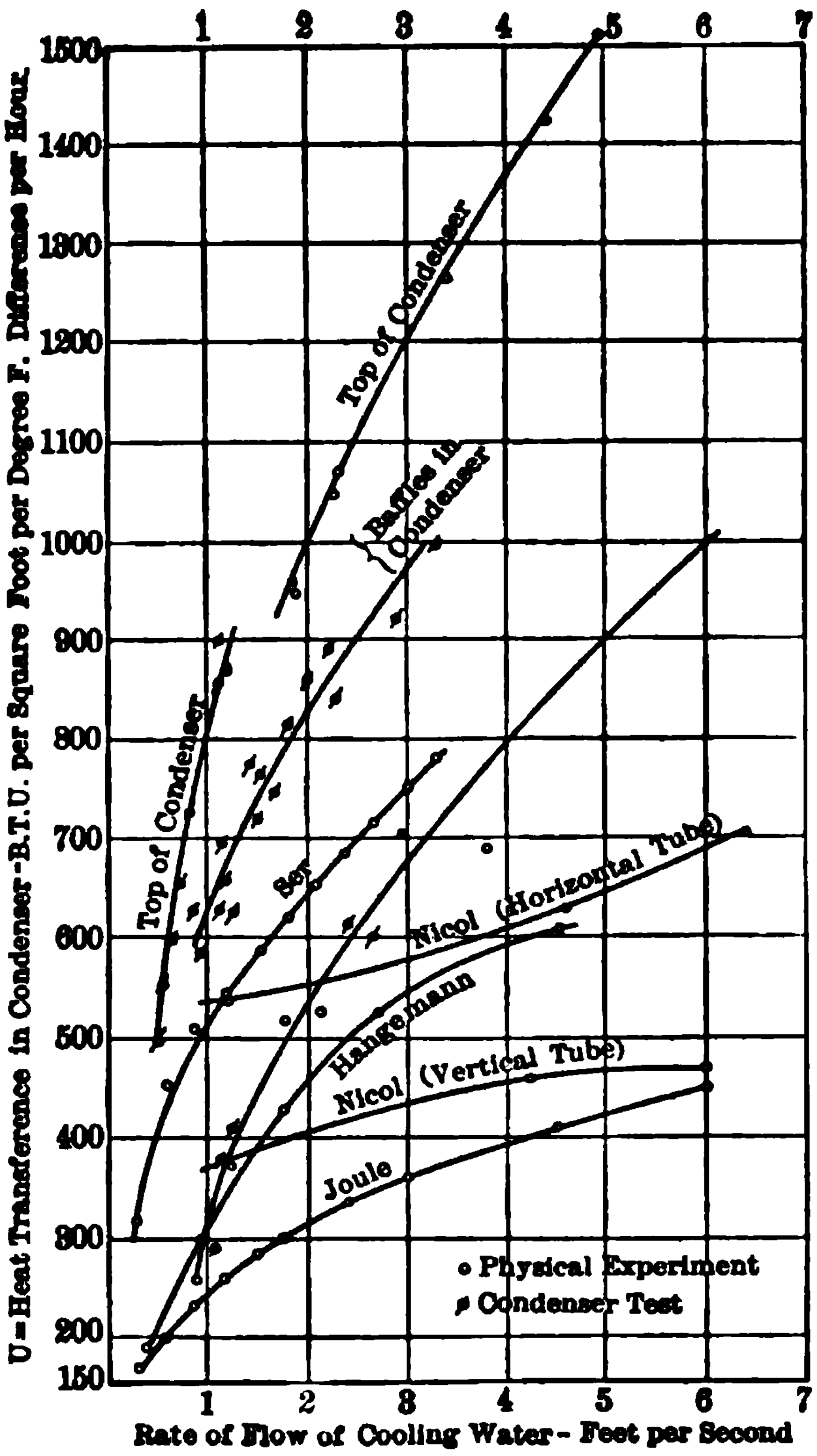


FIG. 216b.

The effect of thickness, material, etc., of condenser tubes is so small in the ultimate result and the choice and arrangement are so largely determined by practical consideration that they may be neglected.

The value of U increases approximately as the square root of the velocity of the water flowing within the tube, so that *increase in water velocity effects a substantial increase in the heat transmission;* but the resistance encountered by the circulating water increases as the square of the velocity, and the power consumed in pumping the water increases as the third power of the velocity, so that a point is soon reached where the gain on the one hand may be offset by the loss on the other.

A study of a number of installations gave

Old-style surface condenser,

V = 30 to 240 ft. per min., average 90.

Modern dry tube surface condenser,

V = 120 to 360 ft. per min., average 240.

From the curves in Figs. 126b and 126c it will be seen that air is an excellent heat-insulating material; hence, the greater the amount of air entrained with the steam the lower will be the coefficient of heat transmission. The necessity of removing the air as fast as it accumulates is at once apparent.

In the older types of surface condensers the water of condensation from the upper rows of tubes is permitted to fall on the rows immedi-

ately below, the water increasing in volume as it passes the successive banks of tubes until it completely envelops them. The coefficient U varies from 1000 or more in the upper row to less than 50 in the lower, giving a *mean* value of approximately 250 to 350 for the entire surface. In estimating the extent of cooling surface for a condenser of this type an average figure for plain brass tubes with water velocities of 50 to 100 feet per minute is $U = 250$. For a velocity of 100 to 240 feet per minute U may be taken 50 per cent greater than these figures. When the tubes are clean a much higher value may be taken, but a liberal factor is usually allowed for possible variation in the condition of operation.

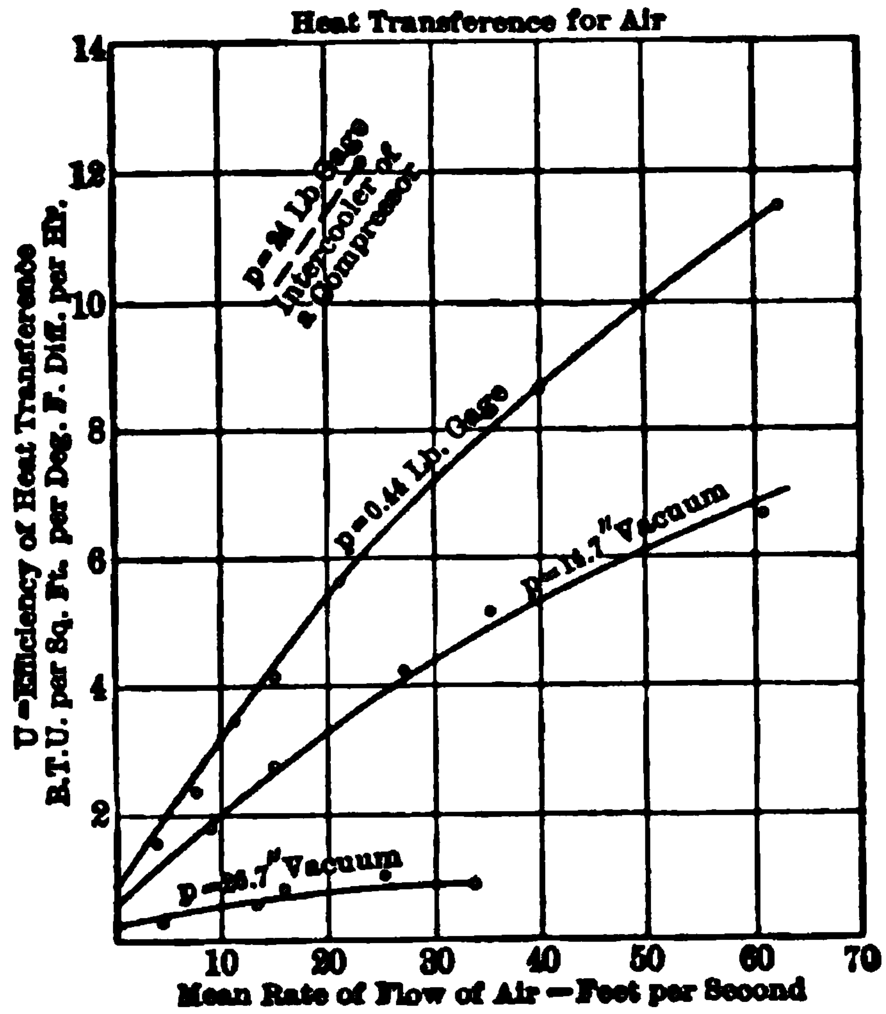


Fig. 216c.

In the modern *dry-tube* surface condenser, designed along the lines of the one described in paragraph 221, in which the water of condensation is withdrawn as rapidly as it is formed, *mean* values of $U = 800$ to 900 are not unusual. In estimating the extent of cooling surface for condensers of this type an average value of U is 600 with water velocities of 4 to 5 feet per second.

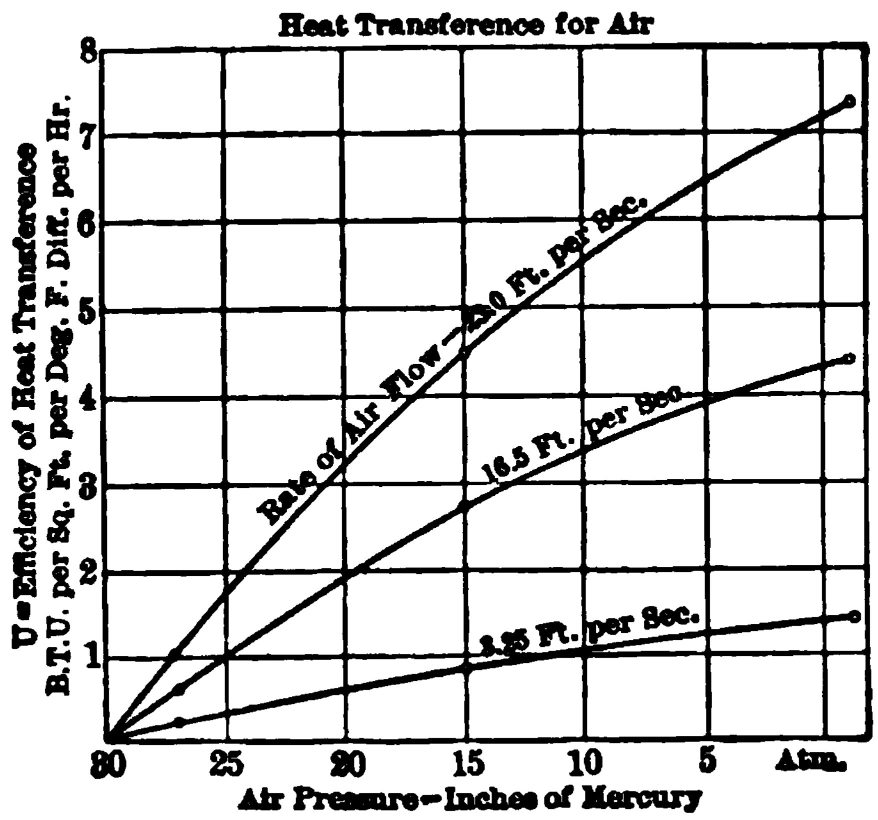


Fig. 216d.

Example: Standard Type of Surface Condenser:—Required the number of square feet of cooling surface per I.H.P. necessary to condense the steam from an engine operating under the following conditions: Engine uses 20 pounds of steam per I.H.P.-hour, vacuum 26 inches with barometer at 30; temperature of cooling water at 60 degrees.

Here $\lambda = 1115$ and $T_s = 126$ (from steam tables),

$$T_0 = 60,$$

$$T_1 = T_s - 10 = 116.$$

In this type of condenser average practice gives a temperature difference of approximately 10 degrees between the temperature of the hot well and that corresponding to the degree of vacuum.

$$T_2 = T_s - 15 = 101.$$

Any value may be fixed upon for T_2 greater than T_0 and less than T_s. The nearer T_2 is to T_0 the greater must be the quantity of circulating water per unit of time for a given rate of condensation. On the other hand, the nearer T_2 is to T_s the less is the mean temperature difference d and hence the greater must be the cooling surface for a given coefficient of heat transmission. When water is cheap and the head pumped against is small T_2 should be given a lower value than when water is costly and the discharge head is large. Average engine practice, with conditions as stated, gives T_2 a value of approximately 15 degrees less than that corresponding to the degree of vacuum.

The logarithmic mean is, equation (118),

$$d = \frac{101 - 60}{\log_e \dfrac{126 - 160}{126 - 101}} = 42.4.$$

The arithmetic mean gives

$$d = 126 - \frac{60 + 101}{2} = 45.5.$$

Substitute the value of d in equation (92) and assume $U = 250$, the figure commonly used for this type of condenser.

$$S = \frac{20\,(1115 - 116 + 32)}{250 \times 42.4} = 1.94,$$

or say two square feet per I.H.P. of engine.

Surface condensers of this type are ordinarily rated on a basis of two square feet per I.H.P.

Example: Dry-tube Multi-flow Surface Condenser: — Required the number of square feet of cooling surface per kilowatt necessary to condense the steam from a steam turbine operating under the following conditions: Turbine uses 15 pounds of steam per kw.-hour; vacuum 28.5 inches, referred to 30-inch barometer; temperature of cooling water 70 degrees.

Here $$\lambda = .9 \times 1100 = 990.$$

The total heat of dry steam corresponding to an absolute pressure of 1.5 inches is 1100, but in the case of high vacuum turbine practice the steam entering the condenser is far from being dry, the quality varying from 0.80 to 0.95, depending upon the quality of the steam at admission. An average figure is 0.9.

$$T_s = 92, \qquad T_0 = 70, \qquad T_1 = T_s - 4 = 88.$$

In this type of condenser the hot-well temperature varies from $T_1 = T_s$ to $T_1 = T_s - 8$.

$$T_2 = T_s - 5 = 87.$$

In this type T_2 varies from $T_2 = T_s$ to $T_2 = T_s - 10$.

$$d = \frac{87 - 70}{\log \frac{90 - 70}{92 - 87}} = 11.5.$$

Arithmetic mean gives $d = 92 - \frac{70 + 87}{2} = 13.5$.

Substitute the value of d in equation (92) and assume $U = 600$, the figure commonly used for this type of condenser.

$$S = \frac{15\,(990 - 88 + 32)}{600 \times 11.5} = 2.02,$$

or say 2 square feet per kw. of generator. There is no standard rating of surface condenser for steam turbine work because of the wide variation in operating conditions. A study of a number of modern installations gives

1 to 2.5 sq. ft. per kw. for large turbo-generators using dry tube surface condensers.

2.5 to 4 sq. ft. per kw. for small turbo-generators using standard surface condensers.

Professor Weighton found from his experiments that a surface condenser constructed on the lines of the one described in paragraph 221 in conjunction with dry-air pumps, was capable of condensing 20 pounds of steam per square foot of surface per hour and maintained a vacuum of 28½ inches (referred to a 30-inch barometer), and this with a cooling-water consumption of 24 pounds per pound of condensed steam; with an inlet temperature of 50 degrees F. a condensation of 35 pounds of steam per hour per square foot of cooling surface was effected at a ratio of 28 pounds of cooling water per pound of steam, the vacuum remaining 28½ inches. See Fig. 216. (*Engineering Record*, May 19, 1906, p. 615.)

EXAMPLES OF MODERN CONDENSER PROPORTIONS.

Name of Station.	Size of Turbo-Generators.	Sq. Ft. of Condenser Surface.	Sq. Ft. of Surface per Kw.
Commonwealth Edison Co.:			
Northwest Station	20,000	32,000	1.60
Quarry Street	14,000	25,000	1.79
Fisk Street	12,000	25,000	2.08
*59th St., Interborough, N. Y.	15,000	25,000	1.67
Metropolitan St. Ry., Kansas City	10,000	22,000	2.20

* Combined Engine and Low-pressure Turbine.

SURFACE CONDENSER AIR PUMPS. — See paragraphs 284–291.

TABLE 52.

SQUARE FEET OF COOLING SURFACE NECESSARY TO CONDENSE AND COOL ONE POUND OF STEAM PER MINUTE.

(Barometer 29.92.)

Temp. of Injection Water.	Vacuum 24". Temperature of Steam 141°.					Temp. of Injection Water.	Vacuum 25". Temperature of Steam 134°.				
	Temperature of Hot Well.						Temperature of Hot Well.				
	110	115	120	125	130		105	110	115	120	125
40	3.1	3.3	3.5	3.7	4.0	40	3.3	3.5	3.7	4.0	4.4
50	3.3	3.5	3.7	4.0	4.3	50	3.5	3.7	4.0	4.3	4.7
60	3.7	3.8	4.0	4.2	4.6	60	3.8	4.0	4.3	4.6	5.1
70	3.9	4.1	4.3	4.6	5.0	70	4.1	4.4	4.7	5.1	5.5
80	4.1	4.4	4.7	5.0	5.5	80	4.5	4.8	5.2	5.6	6.1
90	4.4	4.8	5.1	5.5	6.0	90	5.1	5.4	5.8	6.2	6.9

Temp. of Injection Water.	Vacuum 26". Temperature of Steam 125°.					Temp. of Injection Water.	Vacuum 27". Temperature of Steam 114°.				
	Temperature of Hot Well.						Temperature of Hot Well.				
	100	105	110	115			90	95	100	105	
40	3.6	3.9	4.2	4.6	......	40	4.1	4.4	4.7	5.1	......
50	3.8	4.2	4.6	5.0	......	50	4.4	4.7	5.1	5.6	......
60	4.2	4.6	5.0	5.4	......	60	4.8	5.2	5.6	6.2	......
70	4.6	5.1	5.4	6.0	......	70	5.4	5.8	6.3	7.0	......
80	5.1	5.6	6.1	6.7	......	80	......	6.7	7.2	8.0	......

Temp. of Injection Water.	Vacuum 28". Temperature of Steam 100°.					Temp. of Injection Water.	Vacuum 29". Temperature of Steam 77°.				
	Temperature of Hot Well.						Temperature of Hot Well.				
	75	80	85	90			60	65	70		
40	4.6	4.9	5.3	5.8	......	35	6.3	6.9	7.7	......	......
50	5.0	5.4	5.8	6.4	......	40	6.8	7.4	8.3	......	......
60	5.5	6.1	6.6	7.3	......	45	7.4	8.0	9.0	......	......
70	......	......	7.7	8.5	......	50	......	8.9	9.9	......	.....

Final temperature of injection water assumed to be 10 degrees lower than that of the hot-well.

222. Dry-Air Surface Condensers (Forced Circulation). — Where water is very scarce and the feed supply is reclaimed by condensing the exhaust steam, water-cooled condensers may be prohibitive in cost of operation, even when combined with cooling tower or other water-cool-

ing device, since the latter involves a loss of water approximately equivalent to the amount of steam condensed, due to evaporation.

Under these conditions air cooling has been successfully adopted. In the city of Kalgoorlie, West Australia, an electric station of 2000-horse-power capacity is equipped with air-cooled surface condensers. The condensers have been in use five years (1906), and have given excellent service with very little expense and maintenance. The condenser consists of a large number of narrow chambers constructed of thin corrugated sheet-steel plates spaced ½ inch between centers. Each chamber has 1345 square inches of cooling surface. Fifty-one of these chambers are grouped into a compartment and 15 compartments constitute a section. Each section is equipped with three motor-driven fans 7 feet in diameter and running normally at 320 r.p.m. In all there are six sections, giving a total cooling surface of 45,000 square feet. The steam consumption of the main engines is 16 to 16.5 pounds per I.H.P. hour at rated load. At full load the fans require 130 kilowatts, or approximately 10 per cent of the station output. The average vacuum obtained is about 18 inches throughout the year and ranges from 0 inches on very hot days to 22 inches in cooler weather. The following figures, based on actual observation, show the effect of temperature of the external air on the vacuum when condensing 32,000 pounds of steam per hour (the rated capacity of the condenser).

Temperature External Air, Degrees F.	Vacuum, Inches (referred to 30-Inch Barometer).	Temperature External Air, Degrees F.	Vacuum, Inches (referred to 30-Inch Barometer).
42.8	22	96.8	9.6
50	21.2	100.4	7.6
60.8	20	107.6	3.6
68	18.4	113	0
78.8	16		

Air-Cooled Surface Condensers: Engineering News, Oct., 1902, p. 271; ibid., Vol. 49, p. 203.

223. Quantity of Air for Cooling (Dry-Air Condenser). — The volume of air, under atmospheric conditions, necessary to condense steam to any given temperature may be determined as follows:

Let λ = total heat, above 32 degrees F. of the steam at condenser pressure.

T_s = temperature of the vapor in the condenser.

T_1 = temperature of the condensed steam.

t = temperature of the air entering condenser.

t_1 = temperature of the air leaving condenser.

V = volume of air in cubic feet necessary to condense and cool one pound of steam.

B = specific weight of air under atmospheric conditions.
C = specific heat of air under atmospheric conditions.
d = mean temperature difference between the air and steam.
S = cooling surface in square feet.
U = coefficient of heat transmission, B.T.U. per square foot per degree difference in temperature per hour.

Since the heat absorbed by the air must be equal to the heat given up by the steam, neglecting radiation we have

$$VBC\ (t_1 - t) = \lambda - T_1 + 32, \tag{93}$$

from which

$$V = \frac{\lambda - T_1 + 32}{BC\ (t_1 - t)}. \tag{94}$$

For practical purposes C may be taken as the specific heat of dry air, the error due to this assumption being negligible even if the air is saturated with moisture.

Example: How many cubic feet of air are necessary to condense and cool one pound of steam under the following conditions: Vacuum 20 inches; temperature of entering air, leaving air, and condensed steam, 60, 110, and 140 degrees F. respectively?

Here $\lambda = 1131$ (from steam tables).
$t_1 = 110,\ T_1 = 140,\ t = 60,\ C = 0.2377,\ B = 0.075.$

Substituting these values in equation (94),

$V = \dfrac{1131 - 140 + 32}{0.075 \times 0.2377\ (110 - 60)} = 1150$ cubic feet of air necessary to condense one pound of steam under the given conditions.

The proper area of cooling surface depends upon the value of the coefficient of heat transmission, which varies with the velocity and humidity of the air and character of the cooling surface. Accurate data are not available on this point.

A few experiments made at the Armour Institute of Technology gave values of U = 10 to 25 B.T.U. per hour, per square foot, per degree difference in temperature for air velocities of 500 to 4000 feet per minute for corrugated steel sheeting $\frac{1}{8}$ inch thick. Hence, substituting in equations (94) and (92) we get, for the above example, S = 1.5 square feet of cooling surface per pound of steam condensed per hour for air velocity of 4000 feet per minute, and S = 3.7 square feet for a velocity of 500 feet per minute.

224. Saturated-Air Surface Condensers (Natural Draft). — Fig. 217 shows vertical and horizontal sections of a Pennel saturated-air surface

condenser. The apparatus consists of an upright cylindrical shell containing a number of vertical 4-inch steel tubes through which air is drawn by natural draft. A centrifugal pump circulates about one-half gallon of water per horse power per minute from a cistern below the condenser. The water flowing over the upper tube sheet and then descending the tubes by gravity forms a film over their entire interior surface.

Horizontal Section. Section on AB.

FIG. 217. Pennel Saturated-Air Surface Condenser.

The condensing action is as follows: The current of exhaust steam entering the side of the shell at A is caused by suitable baffle plates to circulate among the tubes, and in condensing gives up its latent heat to the water film, which wholly or partially evaporates, saturating the ascending current of air at its own temperature. The upward current of hot vapor-laden air carries off the heat into the atmosphere. The cooling water which is not evaporated and lost to the atmosphere falls into the cistern below to be again taken up by the circulating pump, the water level in the cistern being kept constant by a float governing a valve on the supply pipe. The non-condensable gases collect at C, where they are removed by the dry-air pump, while the condensed steam is drawn off from the bottom tube sheet by the vacuum pump and discharged into the hot well. An excellent feature of this device is that the film of water on the cooling surface is secured without interference with the ascending air currents and also without the use of sprays through small orifices likely to become clogged with rust or sediment. Where the recovery of the condensed steam is essential and a high vacuum of secondary importance, condensers of this type have proved to be good investments on account of the low first cost.

Table 53 gives the results of a test of a condenser of this type, taking steam from a 30 x 58 x 48 engine running at 45 r.p.m. (*Power*, December, 1903, p. 672; *West. Elect.*, May 19, 1900, p. 323.)

TABLE 53.

TEST OF PENNEL SATURATED-AIR SURFACE CONDENSER.

Duration of trial	9	hours
Average steam pressure at engine by gauge	139.8	pounds
Average vacuum, mercury column	17.5	inches
Average temperature in condenser	123.7	degrees F.
Average temperature of circulating water	116.4	do
Average temperature of city water	52	do
Average temperature of outside air	62	do
Average temperature of saturated air	106	do
Average draft in stack of condenser	1.1	inches
Average humidity of outside air	67	per cent
Average amount of steam condensed per hour	7950	pounds
Average amount of circulating water used per hour	114,660	pounds
Average amount of city water used per hour	3462	pounds
Pounds of city water per pound of steam	2.3	
Pounds of circulating water per pound of steam	14.4	
Average horse power of engine	569.7	
Steam, pounds per I.H.P. per hour	13.95	
Horse power required to run air pumps	10.5	
Horse power required to run circulating pumps	3.0	
Condensing surface, square feet	3900	
Pounds of steam condensed per square foot surface per hour	2038	
Barometer	28.58	inches
Vapor tension corresponding to 123.7 degrees	3.82	inches
Per cent of main engine steam used by auxiliaries	2.38	

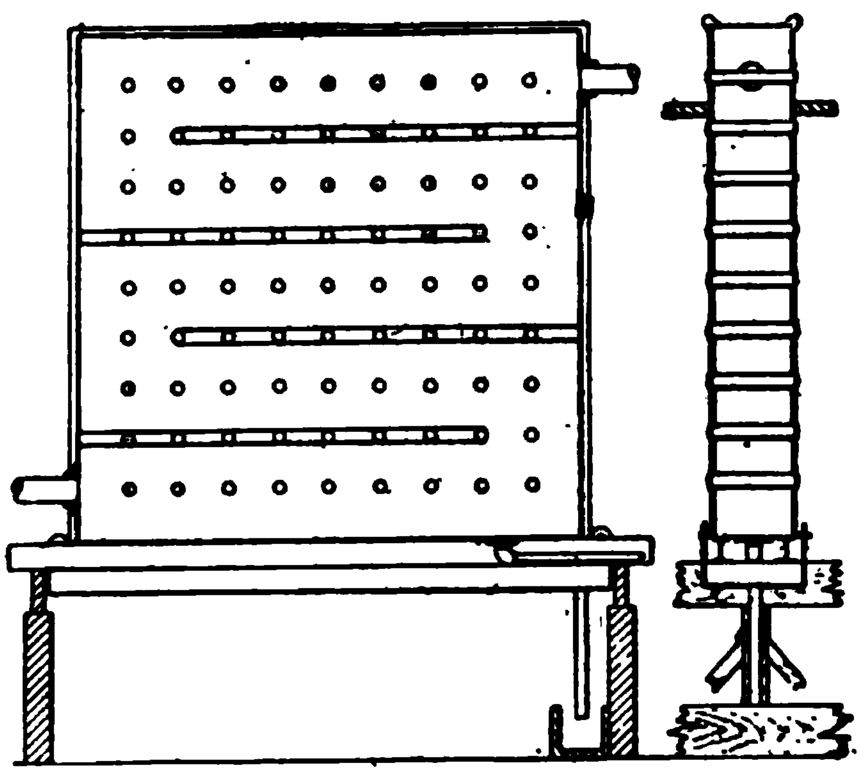


FIG. 218. Pennel Flask Type of Saturated-Air Surface Condenser.

Fig. 218 illustrates the Pennel "flask" type of atmospheric condenser. The exhaust steam enters below and follows the zigzag course bounded by the internal stay channels, condensing as it goes and driving before it the non-condensable gases to the outlet at the top. The condensed steam gravitates to the bottom and thence to the hot well. The top of the flask is trough shaped and causes the cooling water to flow down the sides of the flask in a thin stream. The portion of the cooling water not evaporated collects at the bottom of the flask and flows to the cooling-water reservoir.

225. Evaporative Surface Condensers. — An evaporative surface condenser consists of a number of copper, brass, wrought- or cast-iron tubes arranged horizontally or vertically and connected to manifolds or chambers at each end. The exhaust steam passes through the tubes and a thin film of water is allowed to flow over the external surfaces. The cooling effect is brought about by the evaporation of part of the circulating water, and the general principle of operation is the same as that of the saturated-air condenser described above. Evaporation is sometimes hastened by constructing a flue over the tubes, thereby creating a natural draft, or by means of fans. With horizontal cast-iron tubes and natural draft, vacua from 23 to 27 inches are readily maintained with a cooling surface of approximately eight-tenths square foot per pound of steam condensed per hour. With vertical brass tubes and fan draft 8 pounds of steam per hour per square foot of cooling surface is not an unusual figure. The amount of cooling water evaporated per pound of steam varies from eight-tenths to one pound, depending upon the draft. The power necessary to operate the pumps and fans varies from 1 to 4 per cent of the total output of the plant. For an interesting discussion of evaporative condensers the reader is referred to the admirable article by Oldham in the Proceedings of the Institute of Mechanical Engineers, 1899, and reproduced as a serial in *Engineering* (London), April 28 to June 30, 1899. The following test of a vertical cast iron tube evaporative surface condenser (Table 54) will give some idea of the performance of this type of condenser. This condenser consisted of two rows of 4-inch vertical cast-iron pipes connected at the top by U bends and at the bottom by cast-iron manifolds. A perforated iron trough distributes the water over the center of the bend and causes it to flow in a thin stream over the surface of the tubes. A wet-air pump is used for withdrawing the condensed steam and air. No fan is used for hastening evaporation.*

Evaporative Condensers: Engr., Lond., May 5, 1899, pp. 432, 442, 447; Engineering, May 19, 1899, p. 661, June 2, 1899, p. 721, June 30, 1899, p. 861; Trans. A.S.M.E. 14-696; Power, Sept., 1904, p. 542; Prac. Engr. U.S., June, 1910, p. 346.

226. Location and Arrangement of Condensers. — In the modern power house one sees two general arrangements of condensers and auxiliaries:

1. The independent or subdivided system, in which each engine or turbine is provided with its own condenser, air and circulating pumps.

2. The central system, in which the condensers and auxiliaries are grouped together. Ordinarily one condenser suffices for all engines.

* See end of paragraph 236 for evaporated surface condenser calculations.

TABLE 54.

TEST OF A CAST-IRON, VERTICAL-TUBE, EVAPORATIVE SURFACE CONDENSER, NATURAL DRAFT.

Date	Sept. 12	Sept. 13
Weather	Wet	Fine
Barometer	29.8	29.5
Temperature of air	?	60
Cooling surface, external	272	272
Duration of trial, minutes	99	115
Weight of steam condensed, pounds	800	800
Boiler pressure	60	60
Weight of water in circulation	1830	1830
Weight of fresh water added	600	640
Vacuum in condenser	23.36	24.1
Initial temperature of circulating water	117.5	113.9
Final temperature of circulating water	128.4	125
Temperature of "make up" water	58	58
Temperature of water in hot well	136.5	131.8
Weight of steam condensed per hour, pounds	485	427
Weight of water circulated per hour, pounds	6786	?
Weight of "make-up" water added per hour	364	334
Weight of steam condensed per square foot of cooling surface per hour	1.8	1.54
Weight of "make-up" water per pound of steam condensed, pounds	0.75	0.80

226a. The Independent System.—The condenser is usually placed close to and below the engine so that all condensation may gravitate into it. Figs. 219 and 221 show an application of this system with jet condensers. Here each condenser receives its supply of cooling water from a main injection pipe and discharges into a main overflow pipe. The exhaust pipe leading to the condenser is by-passed through a suitable atmospheric relief valve to a main free exhaust header so that the engine may operate non-condensing in case the vacuum breaks or the condenser is cut out. The chief feature of this arrangement is its flexibility, as each unit is complete in itself and independent of the others. By far the greater number of central stations are equipped with independent condensers.

Occasionally a jet condenser is located on the same level with the engine or even above it, Fig. 222, but such a location should be avoided if possible, as it usually necessitates a larger number of bends and joints in the exhaust pipes than the basement arrangement, and increases the possibility of air leakage. If the exhaust pipe does not drain directly into the condenser, the lowest point in the piping should always be provided with a drip which should be opened when the engine is shut down, as condensation and leakage are apt to fill the pipe with water if the engine stands for any length of time. The end

Fig. 219. Jet Condenser located below Engine-Room Floor.

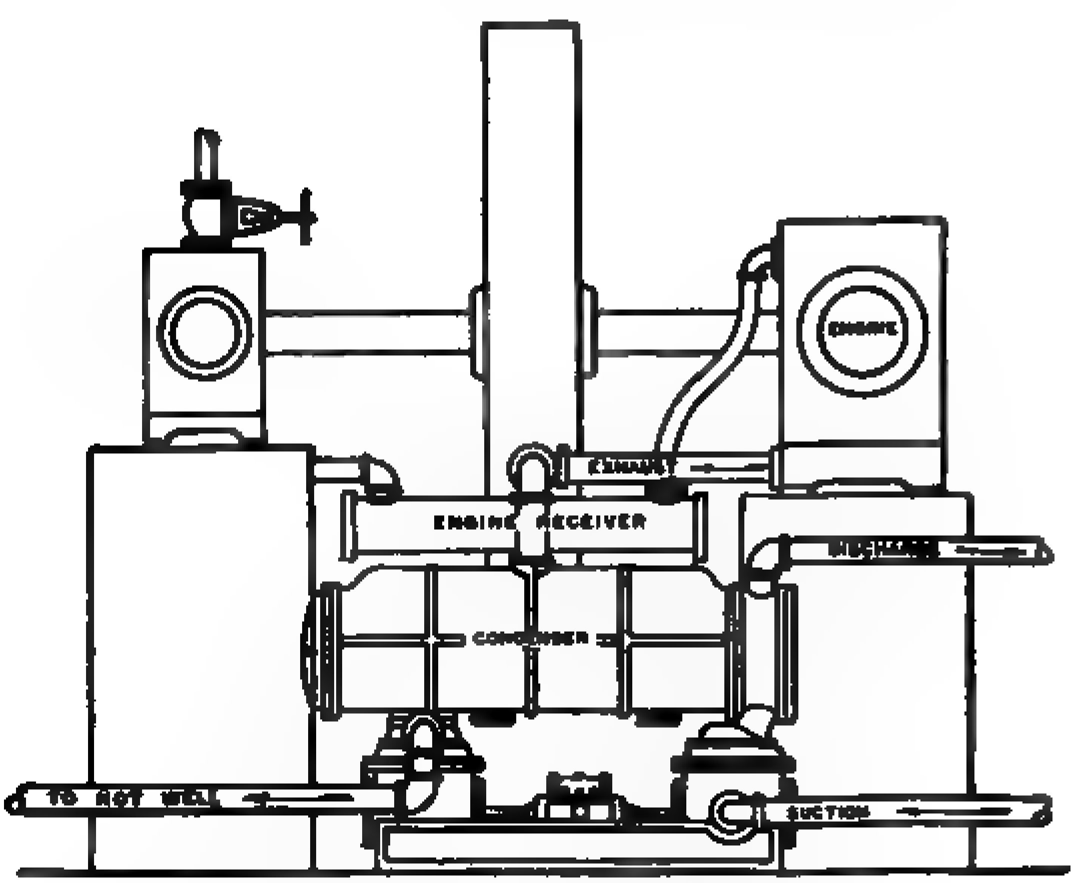


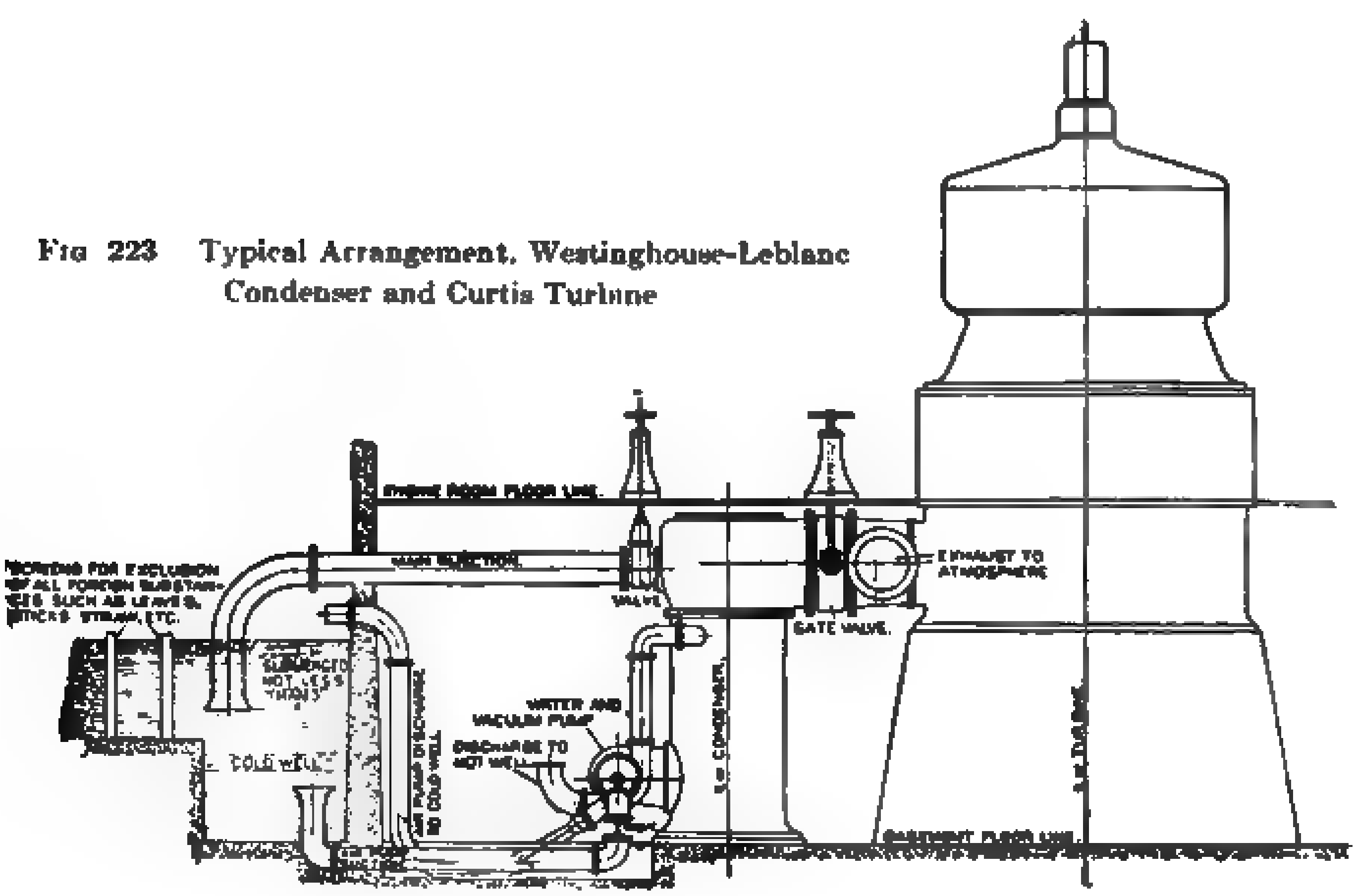
Fig. 220. Surface Condenser located below Engine-Room Floor.

FIG. 221. Surface Condenser, Installed in the Suction Line of a Pumping Engine.

VALVE

FIG. 222. Jet Condenser located above Engine-Room Floor.

FIG 223 Typical Arrangement, Westinghouse-Leblanc Condenser and Curtis Turbine



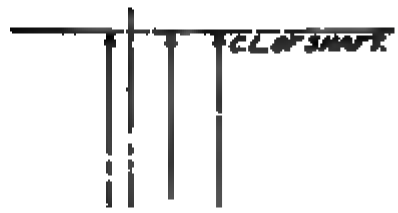


FIG. 224. Elevation of Piping, for Engine and Condenser, Des Moines City Ry. Co.

of the drip should be connected so that water cannot be drawn back through the drip pipe and into the engine cylinder. The length of exhaust pipe and particularly the number of bends between engine and condenser should be kept as small as possible, otherwise the engine may not derive the full benefit of the vacuum in the condenser. A case is recorded where the exhaust piping and appurtenances in connection with a 5000-horse-power engine caused a drop of several inches in vacuum between condenser and exhaust opening of the low-pressure cylinder. (*National Engineer*, December, 1906, p. 10.) The wet-air pump must always be located below the condenser chamber so that the condensation may gravitate to it.

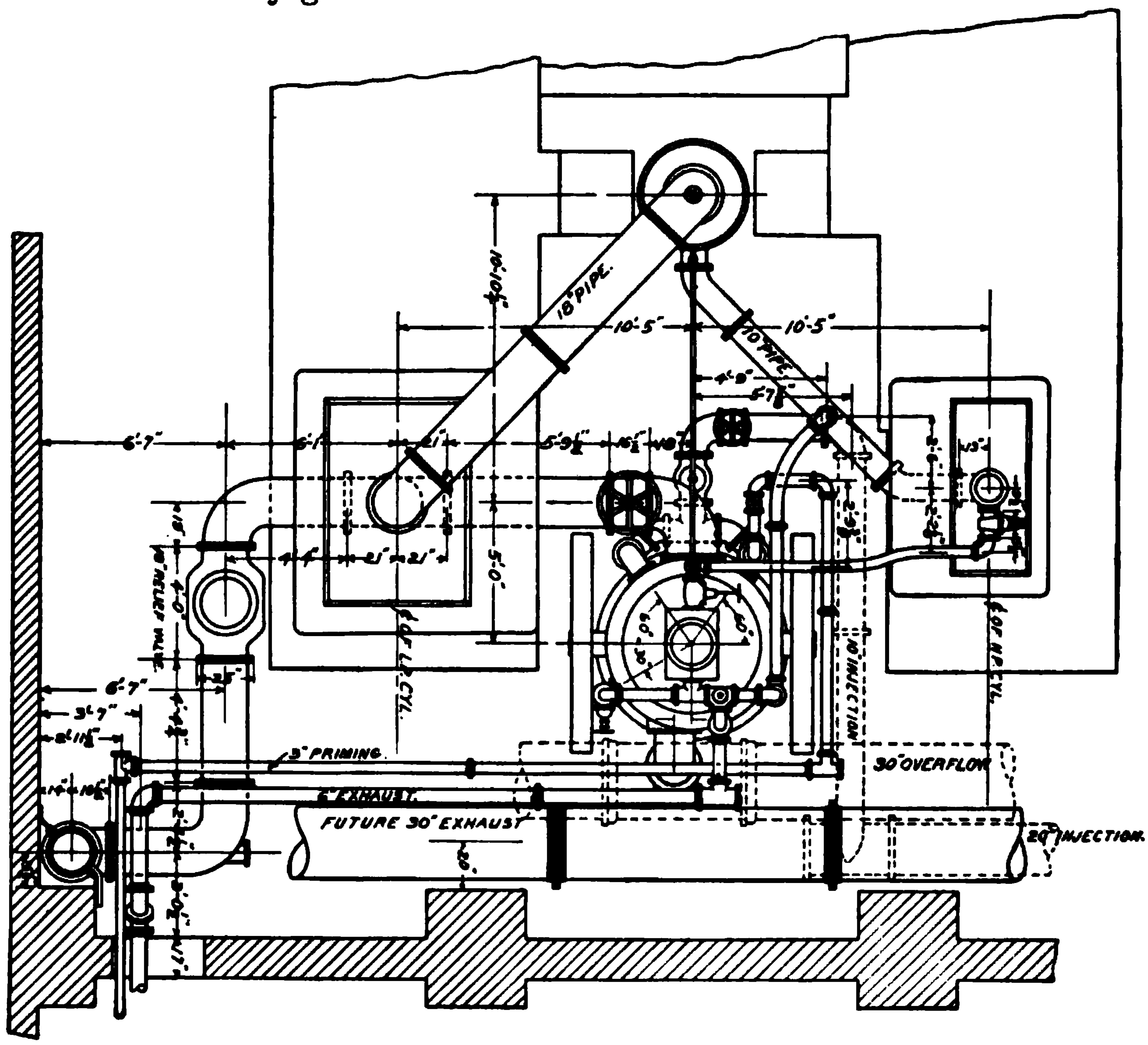


Fig. 225. Plan of Piping for Engine and Condenser, Des Moines City Ry. Co.

Fig. 252 shows a surface condenser installation in connection with a vacuum or primary heater.

Fig. 236 shows an application of a barometric condenser to a vertical engine installation.

Fig. 220 shows the arrangement of a surface condenser with combined air and circulating pump in connection with a horizontal cross

compound engine. The condenser and appurtenances are placed below the engine, thereby permitting the condenser to be closely connected to the engine.

Fig. 221 shows the arrangement of a surface condenser in connection with a pumping engine. The condenser is placed in series with the pump suction.

227. Central Systems. — In the central condensing systems the condenser is located at any convenient point and the exhaust from all the engines piped to it. Any arrangement of condenser and auxiliary machinery may be adopted which will favor the lowest cost of installation and expense of operation. Except where continuity of operation is absolutely essential, only one circulating pump and one air pump are installed. This reduces the number of auxiliary pumps and appliances to a minimum, with a consequent decrease in first cost and maintenance. With properly designed exhaust piping the condenser may be located at a considerable distance from the engine without undue loss of vacuum. At the Cambria Steel Works, Johnstown, Pa., the maximum drop between condenser and engine is only three-quarters of an inch and the distance between them is about 1000 feet.

Central condensers have found great favor in power plants in which the individual units are subjected to extreme variations in load, as in rolling mills. At the works of the Illinois Steel Company, South Chicago, Ill., one condenser takes care of the steam from 15,000 horse power of engines in the rail mill, and another condenses the steam from the 15,000 horse power of engines in the Bessemer steel mill. A notable installation of this system in connection with street-railway work is in the power house of the Northwestern Elevated Company, Chicago, where a single condenser takes care of the exhaust steam of five engines, 11,000 horse power in all. Fig. 226 shows the general arrangement of this installation.

For a comparison of the advantages and disadvantages of the independent and central systems see *Engineering Magazine*, October, 1900, p. 56, *Engineering*, London, June 23, 1899, p. 615, and *Engineering*, July 17, 1903.

Centralization of Steam-Condensing Plant: Eng. Mag., Oct., 1900, p. 56; Iron Age, Jan. 7, 1904; Revue Technique, Feb. 25, 1903.

Five Thousand H.P. Surface-Condensing Plant: Engr., Lond., May 23, 1903.

Aurora & Elgin R.R. Condenser Plant: Engr. Rec., Vol. 47, p. 153.

Condensing Apparatus of Manhattan Elevated Power Plant: Power, Aug., 1903, p. 411.

Interborough R.R. Condenser Plant: St. Ry. Jour., Oct. 8, 1904.

New York Rapid Transit Condenser Plant: Power, June, 1903, p. 283.

Worthington Surface Condensers for Metropolitan Power Station: Power, June, 1901, p. 15.

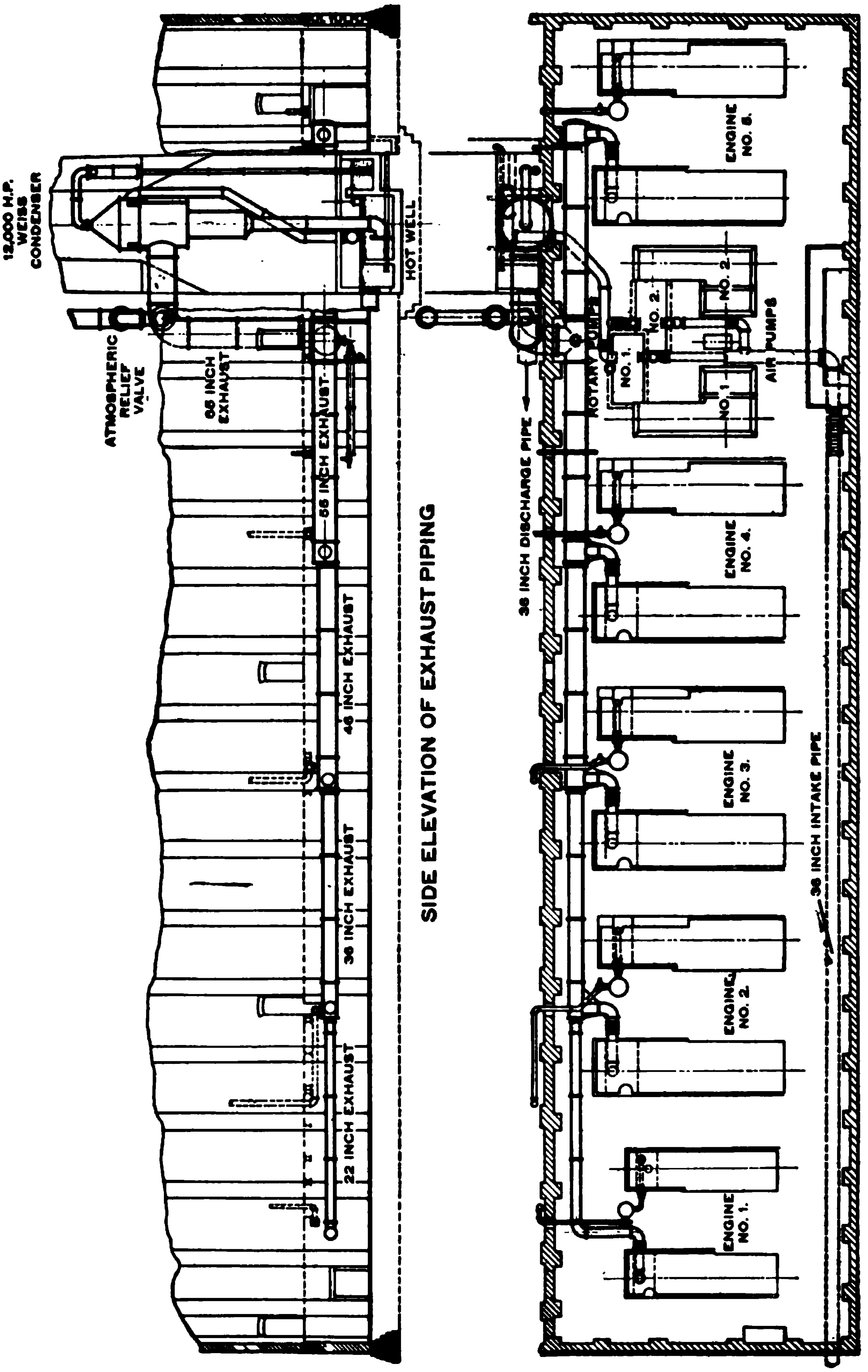


FIG. 226. Condenser and Exhaust Steam Piping at the Northwestern Elevated R.R. Power House, Chicago, Ill.

228. High-vacuum Systems. — The average reciprocating engine gives its best commercial economy at a vacuum of approximately 26 inches (referred to a 30″ barometer), and the ordinary standard jet or surface condenser has been designed to meet this requirement. At the time of the introduction of the steam turbine it was discovered that a very high vacuum would improve turbine economies to an extent hitherto impossible when applied to reciprocating engines. This condition naturally created an era of development among the condenser

FIG. 226a. Condenser Installation, Quincy Point Power Plant of the Old Colony Street Railway Company.

designers. It became evident at once that the old types that were capable of creating a 26″ or 27″ vacuum would require considerable modification to maintain a vacuum of 28″ or 29″. The principal improvement adopted by practically all manufacturers has been to apply a separate dry vacuum pump for the removal of air and non-condensable vapors.

Surface Condensers. — Fig. 227 shows the arrangement advocated by the H. R. Worthington Company. The equipment comprises a surface condenser, a steam-driven centrifugal pump for circulating the cooling water, a steam-driven rotative dry-air pump, and a motor-

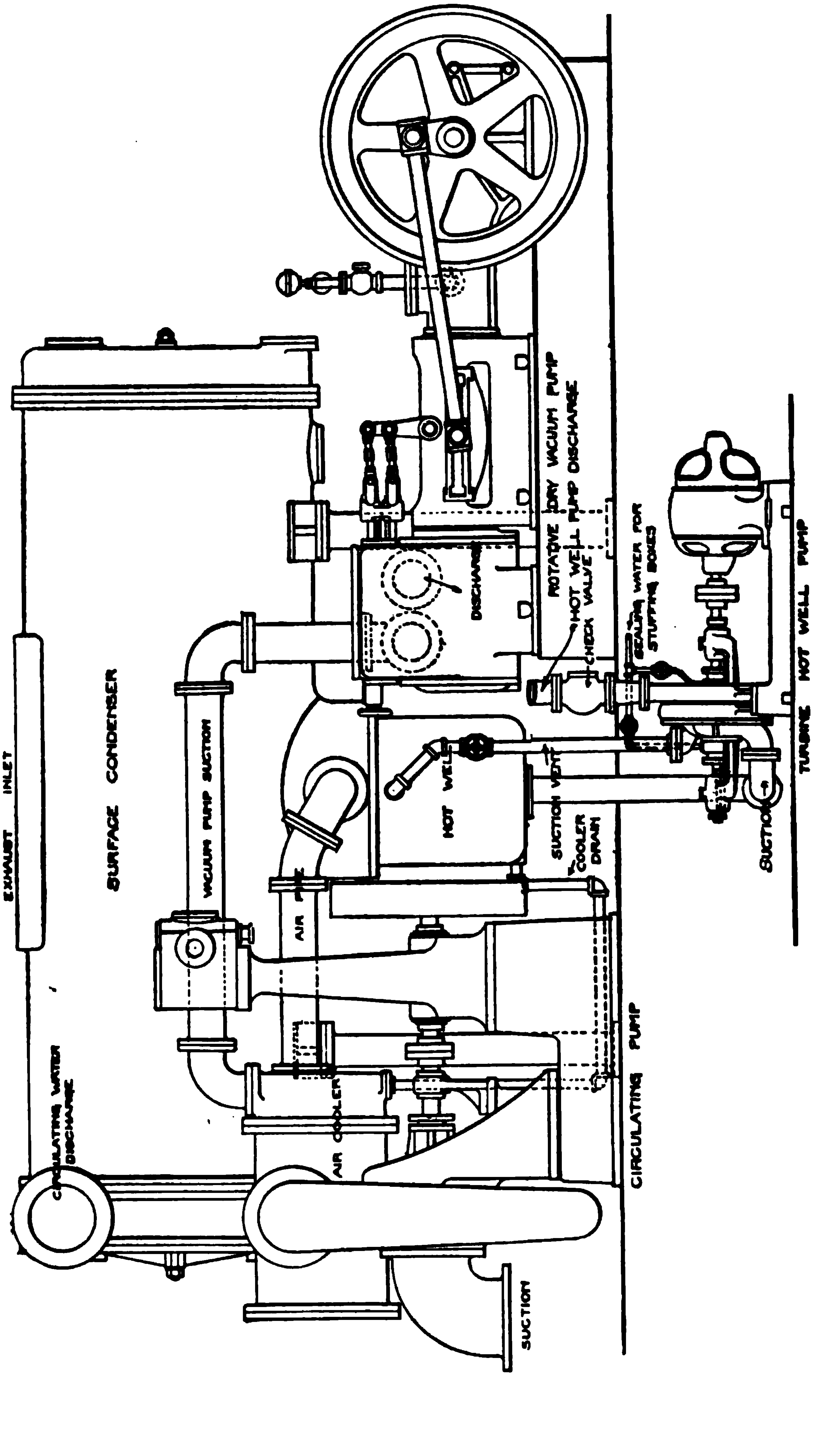


Fig. 227. Worthington High-Vacuum System.

driven centrifugal hot-well pump. The surface condenser is piped direct to the turbine exhaust, only a corrugated copper expansion joint and a tee intervening. A tubular water vapor cooler, which is in reality a small surface condenser, is inserted in the circulating water line between the pump suction and condenser, and serves to arrest all the condensable vapor and thus reduces the volume to be handled by the air pump. All condensation, including that from the air cooler, collects in the hot well, from which it is pumped by a motor-driven circulating pump direct to heater or boiler. Cooling water is handled by a centrifugal pump having both suction and delivery pipes water-sealed, so that the work done by the pump is virtually that of overcoming the

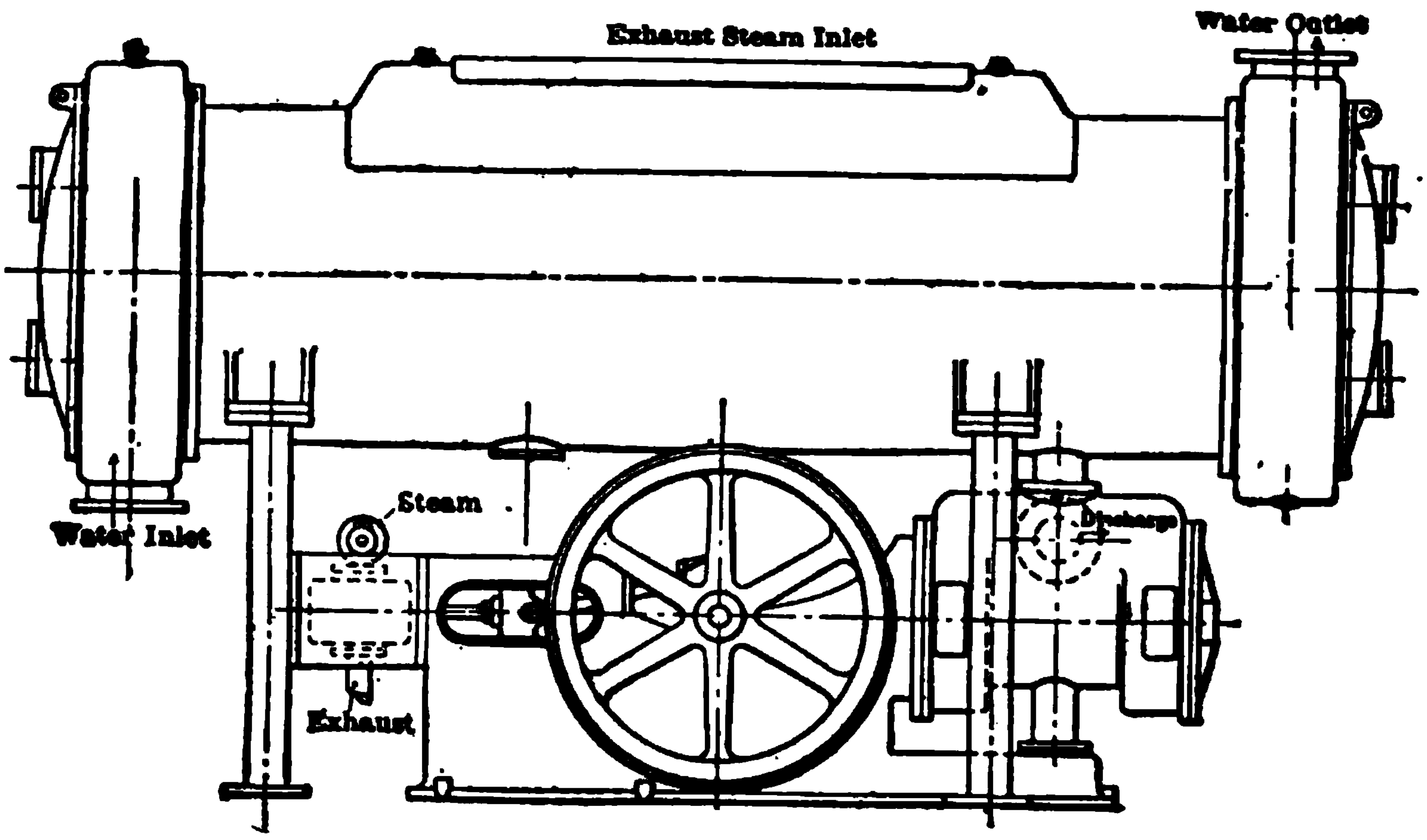


FIG. 228. High-Vacuum System, C. H. Wheeler Co.

fluid friction in the condenser and piping. All valves and stuffing boxes are water-sealed to prevent any possible leakage of air, and the condenser pump cylinder is especially designed to avoid vapor binding. This makes it possible to maintain a vacuum of one-half pound absolute with cooling water at 60 degrees F. In the high-vacuum condenser installation of the Chicago Edison Company the hot-well pump, dry-air pump, and the circulating pump are direct connected to a single-cylinder Corliss engine.

Fig. 228 shows the general arrangement of the C. H. Wheeler Company's high-vacuum condensing outfit. The condensing chamber is shown in section in Fig. 214 and is described in paragraph 219. The wet-air pump is illustrated in Fig. 287 and is described in paragraph 288. No dry-air pump is needed, and the makers guarantee a vacuum within

two inches of absolute under full-load conditions of operating steam turbines.

Fig. 229 shows a section through a Parsons " vacuum augmenter " for increasing the vacuum in a surface condenser. A pipe is led from the bottom of the main condenser to an auxiliary or augmenter having about one-twentieth of the cooling surface of the main condenser. At the point indicated a small steam jet is provided which acts as an ejector and draws out the air and vapor from the condenser and delivers it to the air pump. The water seal prevents the air and vapor from returning to the condenser. With this arrangement, according to tests conducted by Mr. Parsons, if there is a vacuum of 27½ or 28 inches in the condenser, there may be only 26 at the air pump, which, therefore, may be of small

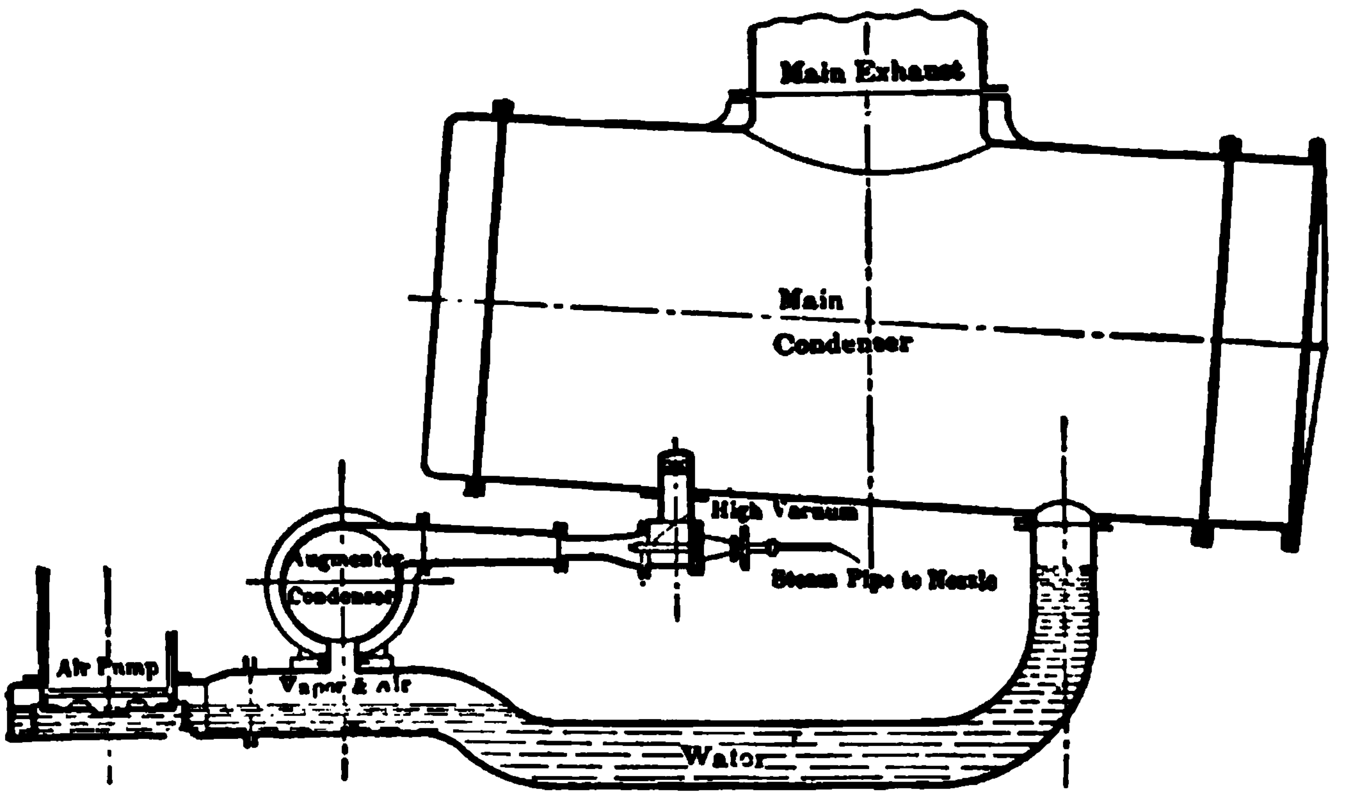


FIG. 229. Parsons Vacuum Augmenter.

size, the jet compressing the air and vapor from the condenser to about one-half of its original volume. The steam jet uses about one and one-half per cent of the steam used by the turbine at full load.

Jet Condensers. — Fig. 229a gives the general details of a Westinghouse-Leblanc multi-jet condenser which, under commercial conditions, has realized vacua within 99 per cent of the ideal. The most striking feature of this system lies in its compactness and simplicity, a 1500-kw equipment being less than 9 feet in height. Referring to Fig. 229a, exhaust steam enters the condenser chamber at the upper left-hand opening and meets the cooling water as it is forced through spray nozzle *C*. The condensed steam and injection water fall to the bottom of the condenser and are removed by centrifugal pump *M*. The non-condensable vapors are withdrawn by valveless rotary air pumps *P*, through suction opening *O*. Referring to section *N-N* through the

air pump it will be seen that this pump consists primarily of a reverse Pelton turbine wheel in conjunction with an ejector. Sealing water is introduced through the branch indicated by dotted outline, into the central chamber *G*, from which it passes through port *H*. It is then caught up by the blades *P* of the Pelton wheel, which is rotated at a suitable speed, and ejected into the discharge cone in the form of thin

Fig. 229a. Westinghouse-Leblanc Multi-jet High-vacuum Condenser System.

sheets having a high velocity. These sheets of water meet the sides of the discharge cone and thus form a series of water pistons, each of which entraps a small pocket of air and forces it out against the atmospheric pressure. In passing through the air pump the sealing water receives practically no increase in temperature, hence the same water may be used over and over again. The air pump rotor and main pump runner are enclosed in a common casing mounted on the same shaft.

This arrangement makes the plant very compact and requires the use of only one motor to drive both pumps. There is a clear passage through the condenser and pump, so that should the pump stop for any reason

Fig. 229b. Tomlinson Type C High-vacuum Jet Condenser.

air rushes into the condenser through the air pump and immediately breaks the vacuum. In starting up the condenser, steam is turned into auxiliary nozzle L, section N-N, for a few moments, thus creating sufficient vacuum to start the regular flow of water through the air pump.

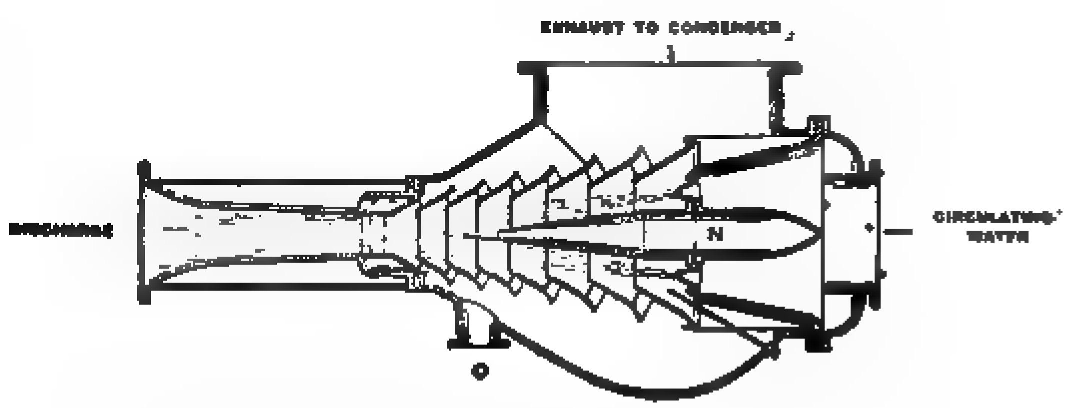


Fig 229c. Section through Condensing Chamber of Körting Multi-jet Condenser. Chamber Capable of Maintaining a Vacuum of 95 per Cent of the Ideal without the Use of Air Pumps.

The pumps require from 1½ to 3 per cent of the power generated by the main engines. Fig. 223 shows an application of a Westinghouse-Leblanc condenser to a Curtis turbine.

229. Power Consumption of Condenser Auxiliaries. — In estimating the cost of producing vacua with the different types of auxiliaries, steam driven, electrically driven, or belted, the power consumption is most conveniently expressed in terms of the equivalent *heat consumption* of the auxiliary in question and not the indicated or developed power. For example, suppose a power plant has a number of 1200-I.H.P. engines direct connected to 800-kilowatt generators and that the engines use 20 pounds of steam per I.H.P. hour at rated load; furthermore suppose the engine driving the air pump (jet condenser) to indicate 24 horse power. Now, it is manifestly incorrect to say that the power consumption of the air pump is equivalent to $\frac{24}{1200} = 2$ per cent of the main engine power unless the engine driving the air pump uses 20 pounds of steam per I.H.P. As a matter of fact the small engine probably uses 30 to 40 pounds or more of steam per I.H.P. hour, and the true power consumption is

$$\frac{24 \times 30}{1200 \times 20} = 3 \text{ per cent, or more.}$$

If the exhaust steam is piped to the condenser, then all of this 3 per cent or more should be charged against the condenser; if the steam is piped to a heater, then only the difference between the heat entering the small engine and that given up to the feed water should be charged against it. For example, suppose the engine in the preceding examples uses 30 pounds of steam per I.H.P. hour when running condensing and 40 pounds when operating non-condensing. Let the initial steam pressure be 150 pounds and feed-water temperature 120 degrees F. when the air pump is running condensing. If the boiler feed is not taken from the hot well, the heat in the exhaust steam is lost so far as the economy of the plant is concerned, and the heat consumption per I.H.P. hour is $30\{1193.6 - (120 - 32)\} = 33{,}168$ B.T.U. This represents the cost, in heat units, of producing the vacuum, and is equivalent to 3 per cent of the main engine output.

If the air pump runs non-condensing and the exhaust steam is piped to the heater, each pound of exhaust steam gives up approximately 950 B.T.U. per hour to the feed water and the temperature of the latter is raised from 120 to 180 degrees F. The heat entering the air pump is $40\{1193.6 - (120 - 32)\} = 44{,}224$ B.T.U. per I.H.P. hour. But $40 \times 950 = 38{,}000$ B.T.U. are returned to the feed water. Hence $44{,}424 - 38{,}000 = 6224$ is the net heat consumption of the air pumps per I.H.P. hour. This corresponds to approximately 0.55 per cent of the main engine output.

In the preceding example suppose the air pump to be motor driven and that it requires 20 electrical horse power per hour. This will be the equivalent of $\frac{20}{0.85 \times 0.90} = 26.2$ I.H.P. of the main engine on the assumption that the efficiency of the small motor is 85 per cent and that of the engine and generator combined 90 per cent. The power required by the air pump will be $26.2 \div 1200 = 2.2$ per cent of the total output.

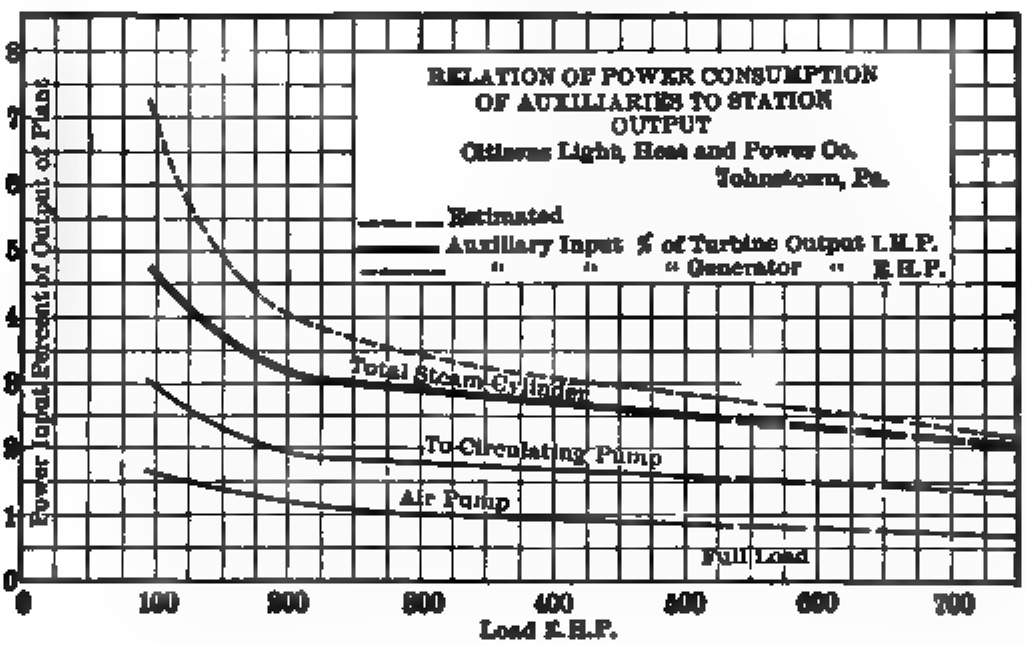


FIG. 230.

In practice the auxiliaries use the equivalent of from 1 to 15 per cent of the main engine or turbine steam, depending upon the size of the plant, character and number of auxiliaries, and the conditions of operation.

FIG. 231.

Table 55 gives the power consumption of the condenser auxiliaries in a number of installations. Fig. 230 shows the relation between the power consumption of the auxiliaries and the total output of the

TABLE 55.

POWER CONSUMPTION OF CONDENSER AUXILIARIES.

Type of Condenser.	Method of Operating Pumps.	Cooling Water, Deg. F.	Vacuum, Inches.	Barometer.	Absolute Pressure, Inches.	Steam Condensed, Lbs. per Hour.	Ratio of Cooling Water to Steam Condensed.	Power Consumed by Pumps.	Per Cent of Total Power Used by Pumps.	Head Pumped Against, Feet.	References.
Hulton, ejector	Electrical	67.6	26.17	30.05	3.88	6,966	68.7	22.5 Kw.	7.1	47.5	Proc. Inst. of Electrical Engineers, Jan. 19, 1905.
Neepsend, surface	do	65.3	27.7	29.9	2.2	29,530	30	37.4 Kw.	2.3	20	
Bromwich, surface	do	51	28.2	29.8	1.6	10,000	50	9.17 Kw.	2.05	15	
Manchester, jet	do	82	26.6	29.5	2.9	35,700	...	40 Kw.	2.2	40	
Surface	......	66.9	28.13	29.93	1.8	95,200	90	102.7 I.H.P.	2.5	..	
Weiss, barometric	Steam	50	27	30	3	128,000	18	80 I.H.P.	1.1	46	
Alberger, surface	do	65	28	30.03	2.03	105,000	81.8	120 I.H.P.	1.5	...	
Surface	do	..	.....	...	.....	10,250	...	......	8.1	....	Nashua Light, Heat, and Power Co.
Tomlinson, barometric	do	40	27.8	30	2.2	70,000	30	45 I.H.P.	1	48	South Side Alley El., Chicago.
Surface	Electrical	30	28	29.5	1.5	32,000	.....	83 Kw.	4.1	30	Los Angeles.

station at different loads for a Parsons steam turbine installation, and Fig. 231 shows a similar relation for a 2000-kilowatt Curtis turbine. (J. R. Bibbins, *Power*, January, 1905.)

Steam vs. Electric Auxiliaries: Engr. U.S., 1902, p. 113; Power, Feb., 1905, p. 90, Sept., 1906, p. 502; St. Ry. Review, March, 1898, p. 184, July, 1899, p. 458.
Centralized Control of Auxiliaries: Engr. U.S., Nov. 1, 1906, p. 782.

230. Cost of Condensers. — The following figures give an idea of the relative costs of the different types of condensers and auxiliaries for a 1000-I.H.P. plant using 20 pounds of steam per I.H.P. hour at rated load, or a total of 20,000 pounds per hour. Vacuum to be maintained, 26 inches, unless otherwise stated; temperature of cooling water, 70 degrees F.; hot-well temperature, 105 to 120 degrees F.; distance between engine exhaust opening and mean level of intake well, 10 feet.

Siphon Condensers.

1 16″ siphon condenser with 6″ centrifugal pump driven by 6″ by 6″ vertical engine $800

Jet Condensers.

1 14″ by 22″ by 24″ jet condenser with single horizontal direct-acting pump 1335

1 16″ by 24″ by 18″ jet condenser with single vertical direct-acting pump 1620

1 14″ by 24″ by 18″ jet condenser with single vertical fly wheel vacuum pump 1770

1 12″ by 17″ by 22″ by 25″ jet condenser, single horizontal direct-acting compound pump 2200

Barometric Condensers.

1 barometric condenser; 10″ by 16″ by 12″ horizontal single-cylinder rotative dry-air pump; 8″ horizontal volute centrifugal pump direct connected to 23-horse-power high-speed engine 2500

1 barometric condenser; 16″ by 16″ dry-air pump direct connected to 9″ by 16″ steam engine; positive rotary pump, for circulating cooling water, belted to above engine 4300

Surface Condensers.

1 surface condenser, 1025 square feet cooling surface, mounted over 7½″ by 14″ by 14″ by 12″ combined air and circulating pump 2100

1 surface condenser, 1025 square feet cooling surface, with 7½″ by 12″ by 12″ horizontal air pump, direct acting, and 6″ centrifugal pump driven by 5″ by 5″ engine 2300

1 surface condenser, 1025 square feet cooling surface; 5″ by 12″ by 10″ Edwards single-cylinder air pump and 6″ centrifugal pump driven by a 5″ by 5″ engine; maximum 28″, referred to 30″ barometer .. 2850

1 surface condenser, 1025 square feet cooling surface; 6″ by 8″ rotative dry-air pump; 6″ by 6″ Edwards wet-air pump and 6″ centrifugal pump driven by 5″ by 5″ engine; maximum vacuum 29″, referred to 30″ barometer (temp. cooling water 50 deg. F.) 3500

In general the cost of complete condensing equipments installed and ready for operation will approximate as follows:

	Cost per Kilowatt of Main Generating Unit.
Siphon condensers without air pump	$2.00 to $ 3.00
Jet condensers	3.00 to 4.50
Barometric condensers with dry-air pump	4.00 to 6.00
Surface condensers for 26-inch vacuum	3.50 to 5.00
High-vacuum surface condensers	3.50 to 10.00

The curve in Fig. 232 shows the relative costs of complete surface condensing plants for steam turbines to maintain the vacua indicated. It will be noted how much more expensive a high-vacuum plant is than one designed for moderate vacua. Thus a 27-inch plant costs 25 per cent more than a 26-inch plant, and a 28.5-inch plant costs twice as much. (J. R. Bibbins, *Power*, January, 1905.)

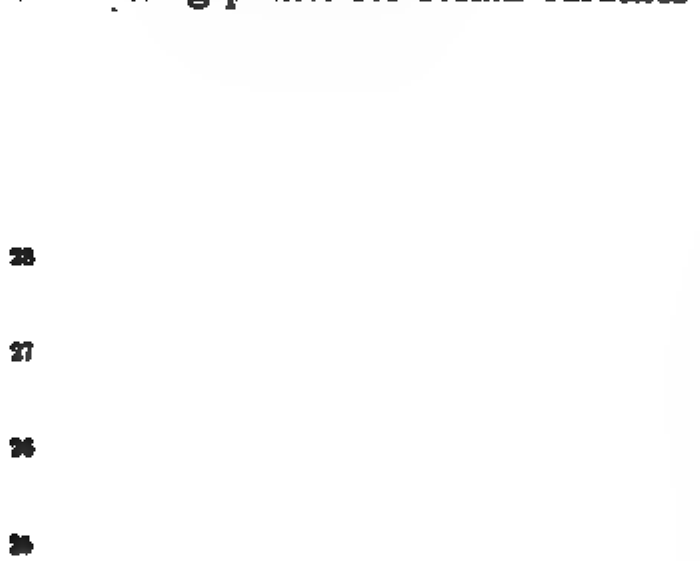


Fig. 232.

The real cost of a condensing plant, however, is not limited to the cost of condensing auxiliaries and piping, but should include all other costs necessitated by the use of the condensing plant, including cost of extra building space, foundations and the like, and the attending fixed charges.

231. Most Economical Vacuum.* — The load factor, or the ratio of the actual yearly load to the rated yearly capacity, has a marked influence on the degree of vacuum best suited for a given installation, since the fixed charges go on whether the plant is running or not, while the gain due to the higher vacuum is realized only when the engines are operating. The higher the load factor the greater is the amount of power produced, the longer does the apparatus operate at best efficiency, the lower the ratio of fixed charges to total operating expenses, and consequently the lower the cost of power per unit.

The load factor for electric-lighting stations is invariably low and seldom exceeds 25 per cent, with an average not far from 18 per cent. In street-railway work it is higher and averages about 30 per cent. In manufacturing plants the load factor varies considerably, but as a rule is somewhat higher than in either of the above cases. Tables 56 and 57 (*Power*, December, 1906, p. 769) show the most economical vacua for different load factors for plants of 1000 kilowatts capacity with

* See also, Elec'n, Lond., Jan. 14, 1910.

conditions as stated. From the tables it would seem at first glance that, except where coal is expensive, all the plants with low factors, 10 per cent and under, ought to be run non-condensing. This is true for "one-engine" installations, but not necessarily so where there are a number of engines or turbines. In the latter case higher economy may be effected by providing only a portion of the engines with condensing equipment. The engine carrying the continuous or day load should operate condensing, and the non-condensing engine should carry the peak load. In order that any of the units may be used for the day work, all engines could be connected to the condenser, but only those carrying the day load should be operated condensing. Each installation, of course, must be considered separately and due weight given to the various factors entering into the problem. For an excellent article on the subject see "Condensers for Steam Engines and Turbines," *Power*, December, 1906, p. 769, and the *Engineer*, London, April 13, 1906, p. 381.

232. Choice of Condensers. — The proper selection of a condenser for a proposed installation depends upon the conditions under which the plant is to be operated. When there is a plentiful and cheap supply of good condensing water suitable for boiler feed, and extremely high vacua are not essential, some type of jet condenser will generally be found most desirable. If overhead room permits, a siphon or barometric condenser will probably be most suitable and least expensive.

TABLE 56.

MOST ECONOMICAL VACUUM FOR STEAM TURBINES.

Vacuum referred to 30-Inch Barometer.

Load Factor, per Cent.	Cost of Coal, Dollars per Ton.									
	$1.50		$2.00		$2.50		$3.00		$3.50	
	A	B	A	B	A	B	A	B	A	B
5	N.C.	N.C.	N.C.	N.C.	18	N.C.	20	N.C.	22	N.C.
10	20	N.C.	23	N.C.	25	N.C.	26.5	20	27	22
15	24	17	26.5	20	27	22	27.5	24	27.7	25.8
20	26.5	20	27.3	23	27.6	25.5	27.8	27	27.9	27.5
30	27.5	24	27.8	27	28	27.6	28.1	27.8	28.2	28
50	28	27.6	28.2	27.9	28.3	28	28.4	28	28.5	28

A. Surface-condensing plant; cost $6 per kilowatt of main generator. Fixed charges 12 per cent. Cost of water not included. Rated capacity of generator, 1000 kilowatts.

B. Surface-condensing plant, including cooling towers and extra cost of land, etc.; cost $10 per kilowatt for 26-inch plant, increasing to $14 per kilowatt for 28.5-inch plant. Fixed charges 12 per cent. No charge for water. Rated capacity of generator, 1000 kilowatts.

TABLE 57.

MOST ECONOMICAL VACUUM FOR RECIPROCATING ENGINES.

Vacuum referred to 30-Inch Barometer.

Load Factor, per Cent.	Cost of Coal, Dollars per Ton.									
	$1.50		$2.00		$2.50		$3.00		$3.50	
	A	B	A	B	A	B	A	B	A	B
10	N.C.	N.C.	15	N.C.	18	N.C.	20	N.C.	22	N.C.
15	16	N.C.	20	N.C.	22	N.C.	22.5	16	24	20
20	22.5	N.C.	23	N.C.	23.5	20	24.5	21	25	22
30	24	16	24.5	21	25.5	22	26.4	23	26.8	24
50	25.5	22	26.7	23.5	27.2	23.5	27.5	26.3	27.7	27

A. Surface-condensing plant; cost $7 per kilowatt of main generator. Fixed charge 12 per cent. Cost of water not included. Rated capacity of generator, 1000 kilowatts.

B. Surface-condensing plant, including cooling towers and extra cost of land, etc.; cost $11 per kilowatt for 26-inch plant, increasing to $13 per kilowatt for 27.5-inch plant. Other conditions as in A.

Where there is a plentiful supply of good water for boiler feed but the water which must be used for cooling purposes is very dirty the siphon condenser is preferable to the barometric form. A surface condenser may be used in the latter case if the condensing water is not so dirty as to seriously impair the efficiency by coating the tubes with sediment, and boiler feed water is scarce.

The air-cooled surface condenser is employed only where water of any kind is scarce.

For very high vacua in connection with steam turbine work the surface condenser is almost universally adopted, although the barometric condenser in connection with dry-air pumps is finding favor with many engineers.

In selecting the type of condenser and auxiliaries due weight must be given to the load factor, cost of coal, water, land, building, interest, depreciation and the like, as outlined in the preceding paragraph.

233. Water-Cooling Systems. — When an ample supply of cooling water is unobtainable for natural or economic reasons, the circulating water may be used over and over again by employing suitable cooling devices. The three most common in practice are

1. The simple cooling pond or tank.
2. The spray fountain.
3. The cooling tower.

233a. Cooling Pond. — The water is cooled partly by radiation and conduction but principally by evaporation. The air is seldom saturated normally, and its capacity for absorbing moisture is increased on account of its temperature being raised by contact with the warm water and by radiation. The cooling action is independent of the depth of water and varies directly as the surface, the amount of heat dissipated for each square foot depending upon the temperature of the water, the relative humidity, and the velocity of the air currents. Results of tests are very discordant.

Box in his treatise on Heat states that the pond surface should approximate 210 square feet per nominal horse power for an engine working twenty-four hours a day. (Treatise on Heat, Box, p. 152.)

If the engine works only twelve hours per day, the area may be reduced to 105 square feet per horse power, because the water will cool during the night, but in that case the depth should be such as to give a capacity of 300 cubic feet per horse power. These figures are based on a reduction in temperature of 122 to 82 degrees F., with air at 52 degrees F. and humidity 85 per cent, the steam consumption per nominal horse power being taken at 62.5 pounds.

Box gives the following formula for the rate of evaporation in perfectly calm air:

$$E = (243 + 3.7\,t)\,(V - v), \tag{95}$$

in which

E = evaporation in grains per square foot per hour.
t = temperature of the water, degrees F.
V = maximum vapor tension in inches of mercury at temperature t.
v = actual vapor tension.

Evaporation is greatly affected by the force of the wind and varies from 2 to 12 times the amount determined from equation (95).

Example: How many pounds of water will be evaporated per square foot per hour from a pond with the temperature of the water and air 80 degrees F.; air perfectly calm; barometric pressure 29.5 inches and relative humidity 70 per cent?

The maximum vapor tension at temperature of 80 degrees is 1.02 inches of mercury. The actual vapor tension will be

$$1.02 \times .70\ (= \text{relative humidity}) = .714.$$

Substitute these values in (95).

$$\begin{aligned} E &= (243 + 3.7 \times 80)\,(1.02 - 0.714) \\ &= 165 \text{ grains per square foot per hour.} \\ &= .023 \text{ pound per square foot per hour.} \end{aligned}$$

If the temperature of the water were 130 degrees F. and that of the surrounding air 80 degrees F., humidity 70 per cent, the evaporation would be

$$
\begin{aligned}
E &= (243 + 3.7 \times 80)(4.5 - 0.714) \\
&= 2040 \text{ grains per square foot per hour} \\
&= 0.291 \text{ pound per square foot per hour.}
\end{aligned}
$$

Here 4.5 = maximum vapor tension, corresponding to a temperature of 130 degrees.

233b. Spray Fountain. — From equation (95) we see that even under the most favorable circumstances an enormous pond surface is necessary. To facilitate evaporation with a view toward reducing the size of the pond, the hot circulating water is sometimes distributed through pipes and discharged through nozzles, falling to the surface of the pond in a spray. The following gives some interesting data con-

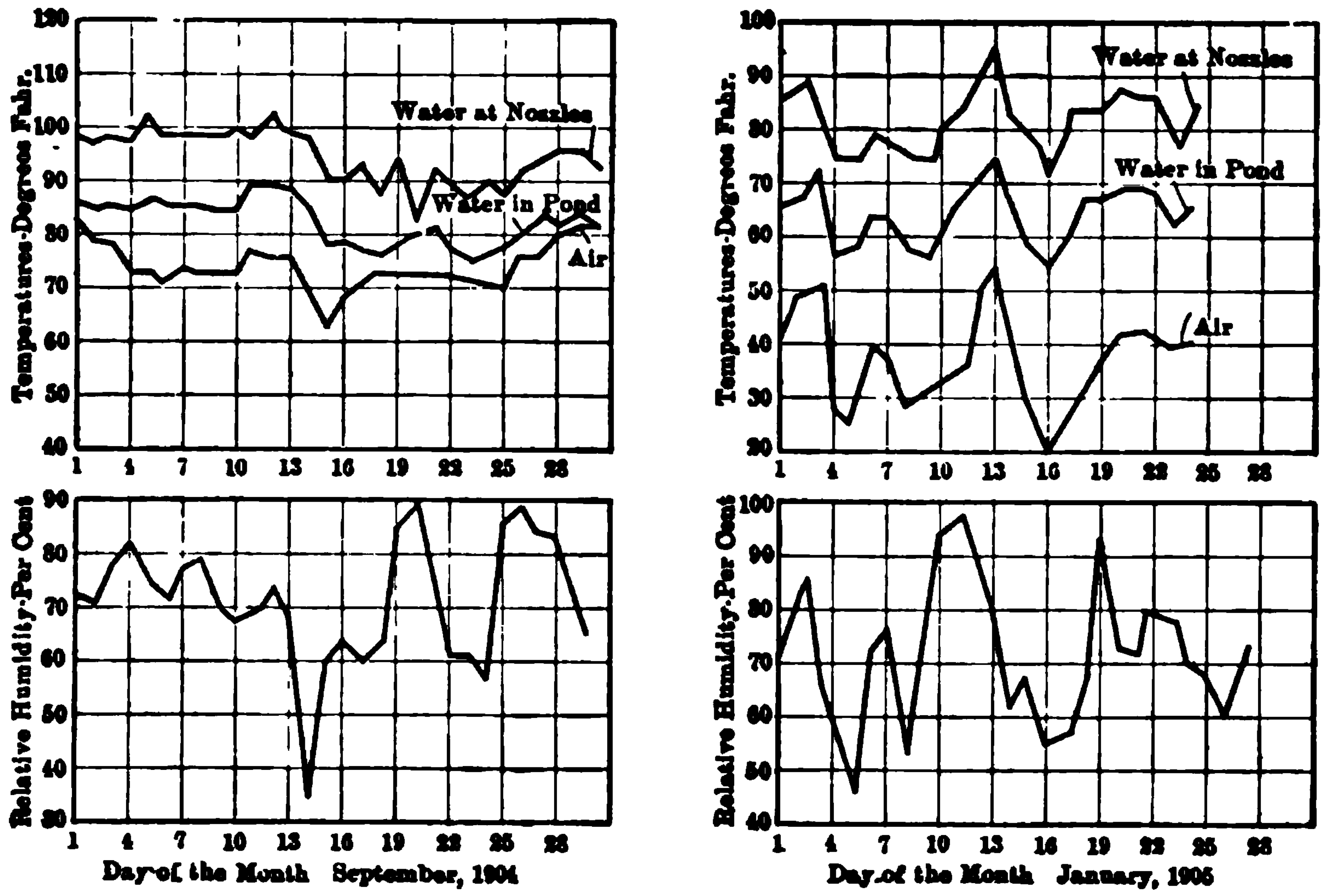


FIG. 233. Curves Showing Performance of Spray Fountain; Chattanooga Electric Company's Power Plant.

cerning the spray fountain installation at the power plant of the Chattanooga Electric Company, Chattanooga, Tenn. (*Street Railway Review*, March 15, 1905.)

Adjoining the power house a pond 150 x 300 feet was excavated to a depth of 4 feet, the level of the water being 8 feet below the condensers.

Circulating water returned from the condensers is distributed through a set of pipes provided with 42 nozzles through which the water is discharged upwards. The rectangle defined by the center lines of the outermost pipes is 98 feet by 125 feet. The pipes are supported on brick piers spaced at intervals of about 20 feet in each direction. The discharge opening of the nozzles is 1¼ inches in diameter, and the interior is provided with a spiral core so that in its passage the water is given a rotary motion, the effect of which is to greatly increase the spraying action. The nozzles, except on the extreme outer lines of piping, are placed in pairs with the axes in a vertical plane at right angles to the center line of the supply pipe, the axis of each nozzle making an angle of 30 degrees with a vertical plane through the center of the supply pipe. The effect of each pair of nozzles is to throw a mass of spray to the height of about 15 feet, which in falling covers an area of 15 x 30 feet.

A dike extending nearly across the pond near one end provides a canal through which the water is conducted to the suction chamber, the object being to draw the supply from distant parts of the pond to give greater time for cooling. The "make-up" water is supplied by wells. The operation of the cooling pond for a warm month and for a cold month is shown in Fig. 233. Readings were taken at three-hour intervals. The pond supplies the circulating water for three 2000-square-feet Worthington surface condensers.

234. Cooling Towers. — A cooling tower consists of a wooden or sheet-iron housing open at the top and bottom and so arranged that the heated cooling water may be elevated to the top and distributed in such a manner that it falls in thin sheets or sprays into a reservoir at the bottom, air at the same time being drawn in at the bottom by natural draft or forced in by a fan. The water gives up its heat to the ascending current of air by evaporation and conduction, the latter, however, being a relatively small factor. If the air supply is dependent entirely upon convection, the system is known as the natural-draft or flue cooling tower; if the air is forced into the tower by fans, it is called a fan cooling tower. The different types vary principally in the method of water distribution. Fig. 234 illustrates the Barnard cooling tower, in which the falling water is broken up by vertically suspended galvanized iron wire cloth mats, causing it to trickle in thin sheets to the bottom. A similar result is brought about in the Worthington tower, Fig. 235, by pieces of terra-cotta pipe 6 inches in diameter and two feet long placed on ends in rows. In the Alberger cooling tower the water trickles down the sides of swamp-cypress boards arranged in honeycomb fashion. In the Jennison cooling tower the

water is divided into a rain of drops, constantly retarded in their fall by a series of perforated 4 x 4 inch galvanized iron trays arranged in horizontal rows and staggered vertically.

With the best forms of cooling towers, under average conditions, the

FIG. 234. Barnard-Wheeler Cooling Tower.

temperature of the circulating water may readily be reduced from 40 to 50 degrees with a loss not exceeding 3 or 4 per cent of the total quantity of water passing through the tower. The power consumed by the fan in a forced-draft apparatus averages 2 per cent of that

developed by the main engines, for the maximum requirements during summer months, and 1½ per cent during the winter.

The location of the tower may be on the engine-room floor, on top of

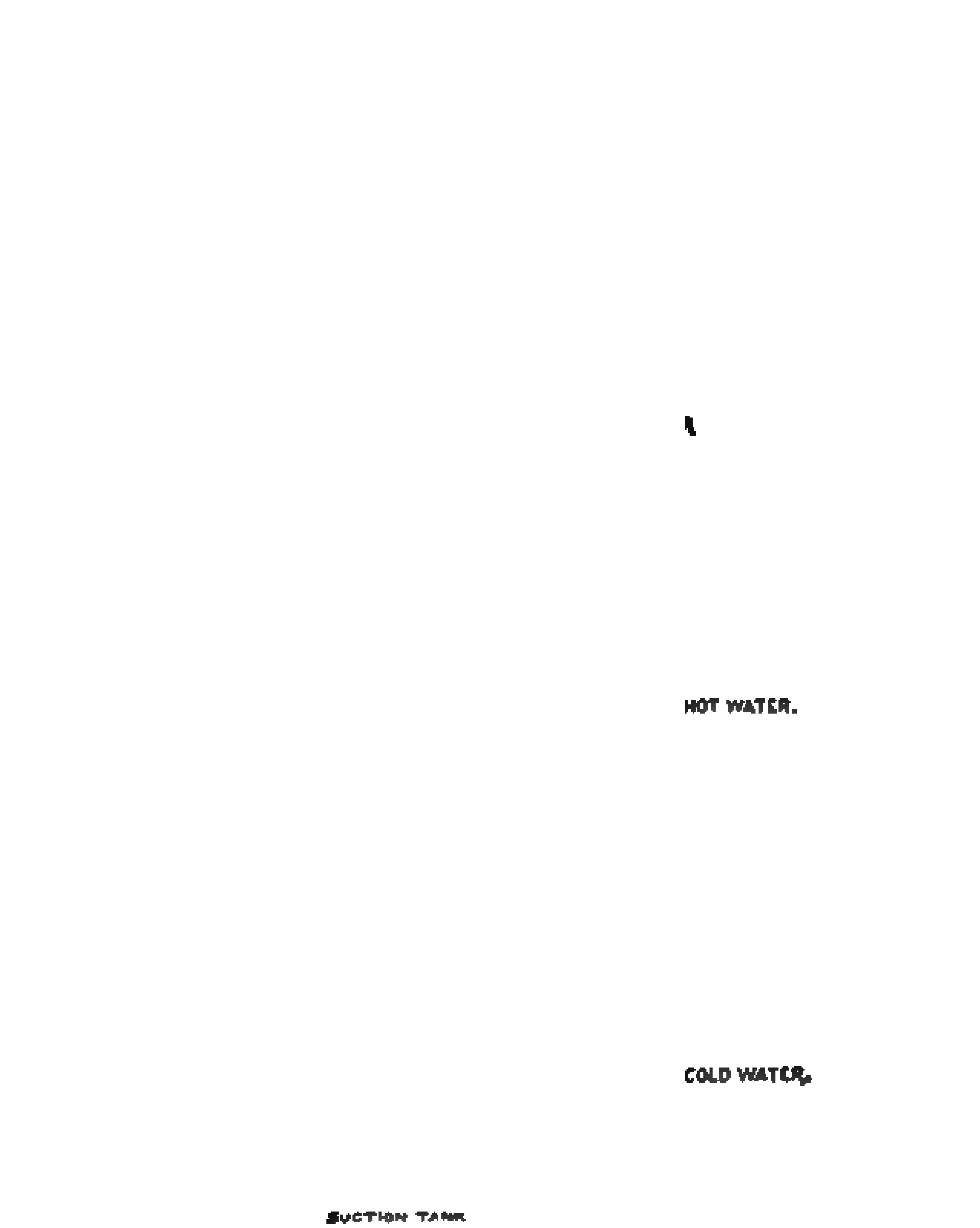


Fig. 235. Worthington Cooling Tower.

the building, or in the yard, the latter being the most adaptable. It may be any reasonable distance from the engine and condenser. Fig. 236 shows a typical installation of Alberger condenser and cooling towers.

Fig. 236. Typical Cooling Tower Installation

235. Parallel Comparison of Fan and Natural-Draft Cooling Towers.

FAN.	NATURAL DRAFT.
	Size.
Small, the forced draft providing sufficient air velocity to effect evaporation. Height limited, because loss from back pressure increases with the height. Tower usually short and of large area.	Large, draft being necessarily small, a larger area must be provided to perform same work. Height is an advantage because the tower operates on the principles of a chimney.
	Power Consumption.
One per cent of station output and upwards, depending upon the type of auxiliaries and the conditions of operation.	None.
	Location.
Inside or outside. Can operate in any location where sufficient head room and air supply are available. Especially adapted to inconvenient locations, as roofs, upper decks, boiler floors, etc.	Outside only, unless exceptionally good draft is obtainable. Preferably in the open where advantage may be taken of prevailing winds.
	Conditions of Atmosphere.
Comparatively little affected by temperature, considerably by humidity, and none by winds.	Largely affected by temperature and humidity and wind. Draft increased by steady winds.
	Conditions of Operation.
More especially adapted for heavy continuous duty the year round, as in rail-plants or mills.	Especially adapted for light summer and heavy winter duty, as in electric-lighting plants.
	First Cost and Cost of Operation.
First cost greater on account of mechanical construction and necessary auxiliaries. Cost of operation dependent upon type of auxiliary and conditions of operation.	First cost small by reason of simplicity and construction. First cost largely dependent upon materials used in interior construction. Cost of operation limited to fixed charges.

236. Water-cooling Calculations. — Air is said to be completely saturated when it contains all the water vapor it can hold without

causing precipitation. If the vapor content is less than that corresponding to complete saturation the air will tend to become saturated by absorbing moisture from surrounding objects. The drier the air the greater will be its affinity for moisture. The necessary latent heat for vaporization is supplied directly by the water producing the vapor or by the surrounding objects in contact with the water. Thus, in the open cooling-tower the water vapor is absorbed from the circulating water, and the heat necessary to effect this vaporization is given up by the water, with a resultant reduction in temperature of the water itself; and in the evaporative surface condenser the vapor is absorbed from the water spray in contact with the tubes, the heat required to effect this vaporization being given up by the steam within the condenser chambers, resulting in condensation of the steam. If the air coming in contact with the water is very dry and at a high temperature the vaporization of the water may be rapid enough to cool the remaining water to a temperature much lower than that of the air. In this case practically all of the cooling is effected by *evaporation*. But when the air is at a low temperature and high relative humidity a considerable amount of heat may be carried away by the air by *conduction*. The *quantity* of air and water necessary to produce a given cooling effect may be determined as follows:

Let H = total amount of heat to be abstracted, B.T.U. per hour.
W = weight of water to be cooled, lbs. per hour.
t_e = temperature of water entering cooling device.
t_l = temperature of water leaving cooling device.
t_0 = temperature of air entering cooling device, ° F.; $T_0 = t_0 + 460$.
t_2 = temperature of air leaving cooling device, ° F.; $T_2 = t_2 + 460$.
p = ordinary atmospheric pressure = 29.92 in. of mercury.
p_a = observed atmospheric pressure, in. of mercury.
p_0 = elastic force of vapor at temperature t_0, in. of mercury.
p_2 = elastic force of vapor at temperature t_2, in. of mercury.
V_0 = volume of air entering the cooling device, cu. ft. per hour, atmospheric conditions.
V_2 = volume of air discharged from the cooling device at temperature t.
d = density of *dry* air, at pressure p and temperature t_0.
h_0 = weight of moisture in 1 cu. ft. of saturated air at temperature t_0, pounds.
h_2 = weight of moisture in 1 cu. ft. of saturated air at temperature t_2, pounds.
z_0 = relative humidity of the air entering the cooling device.
z_2 = relative humidity of the air leaving the cooling device.

TABLE 58.

PROPERTIES OF AIR.

Weights of Air, Vapor of Water, and Saturated Mixtures of Air and Vapor of Different Temperatures, under the Ordinary Atmospheric Pressure of 29.921 Inches of Mercury.

			Elastic Force of Vapor in Inches of Mercury (Regnault).	Mixtures of Air Saturated with Vapor.										Warmed One Degree per B.T.U.	Temperature F.
				Elastic Force of the Air in the Mixture of Air and Vapor in Inches of Mercury.	Weight of a Cubic Foot of the Mixture.			Ratio of Water Vapor to Dry Air.							
					Weight of the Air in Pounds.	Weight of the Vapor in Pounds.	Total Weight of Mixture in Pounds.								
			4	5	6	7	8	9						15	1
			0.044	29.877	.0863	.000079	.086379	.00092						8.7	0
			.074	29.849	.0840	.000130	.084130	.00115						0.0	12
22	.980	.0824	.118	29.803	.0821	.000202	.082302	.00245						1.0	22
32	1.000	.0807	.181	29.740	.0802	.000304	.080504	.00379						1.8	32
42	1.020	.0791	.267	29.654	.0784	.000440	.078840	.00561						2 8	42
52	1.041	.0776	.388	29.533	.0766	.000627	.077227	.00819						3.8	52
60	1.057	.0764	.522	29.399	.0751	.000830	.075252	.01105						4.9	60
62	1.061	.0761	.556	29.365	.0747	.000881	.075581	.01179						5.2	62
70	1.078	.0750	.754	29.182	.0731	.001153	073509	.01780						5.7	70
72	1.082	.0747	.785	29.136	.0727	.001221	.073921	.01860						5.8	72
82	1.102	.0733	1.092	28.829	.0706	001667	.072267	.02361						6.5	82
92	1.122	.0720	1.501	28.420	.0684	.002250	.070717	.03289						7.1	92
100	1.139	.0710	1.929	27.992	.0664	.002848	.069261	.04495						7.8	100
102	1.143	.0707	2.036	27.885	.0659	.002997	.068897	.04547						7 8	102
112	1.163	.0694	2.731	27.190	.0631	.003946	.067042	.06253						8.5	112
122	1.184	.0682	3.621	26.300	.0599	.005142	.065046	.08584						9.1	122
132	1.204	.0671	4.752	25.169	.0564	.006639	.063039	.11771						9.9	132
142	1.224	.0660	6.165	23.756	.0524	.008473	.060873	.16170						0.6	142
152	1.245	.0649	7.930	21.991	.0477	.010716	.058416	.22465	4.45	93.3	.01544	.01654	65.0	60.5	152
162	1.265	.0638	10.099	19.822	.0423	.013415	.055715	.31713	3.15	74.5	.01518	.01656	65 6	60.4	162
172	1.285	.0628	12.758	17.163	.0360	.016682	.052682	.46338	2.16	59.2	.01494	.01658	67 1	60 3	172
182	1.306	.0618	15.960	13.961	.0288	.020536	.049336	.71300	1.402	48.6	.01471	.01687	68.0	59.5	182
192	1.326	.0609	19.828	10.093	.0205	.025142	.045642	1.22643	.815	39.8	.01449	......	68 9		192
202	1.347	.0600	24.450	5.471	.0109	.030545	.041445	2.80230	.357	32.7	.01466	....	68.5	....	202
212	1.367	.0591	29.921	0.000	.0000	.036820	.036820	Infinite	.000	27.1	.01406	....	71 4		212

C = specific heat of dry air at constant pressure, = 0.2375.
S = mean specific heat of water vapor at temperature t_2.
r_2 = heat of vaporization at temperature t_2.

The *pressure* p_3 of the *dry* air in atmospheric air entering the cooling device is

$$p_1 = p_a - p_0 z_0. \tag{96}$$

The *pressure* p_3 of the *dry* air leaving the cooling device is

$$p_3 = p_a - p_2 z_2. \tag{96a}$$

The *weight* w of *dry* air entering the cooling device under atmospheric temperature and pressure, lbs. per hour, is

$$w = \frac{p_1}{p} d V_0. \tag{97}$$

The *weight* w_0 of *moisture* carried into the cooling device by the air, lbs. per hour, is

$$w_0 = h_0 z_0 V_0. \tag{97a}$$

The *volume* of *air leaving* the cooling device is

$$V_2 = V_0 \frac{p_1}{p_3} \cdot \frac{T_2}{T_0}. \tag{98}$$

The *weight* w_2 of *moisture* carried away by the air *discharged* from the cooling device, lbs. per hour, is

$$w_2 = h_2 z_2 V_0. \tag{98a}$$

The *weight* of *circulating water* w_3 *absorbed by the air in passing through the cooling device*, lbs. per hour, is

$$w_3 = w_2 - w_0. \tag{98b}$$

The *heat* H to be *abstracted* from the *circulating water*, B.T.U. per hour, is

$$H = W(t_l - t_e). \tag{99}$$

The heat is dissipated, by the cooling process, in raising the temperature of the air and vapor entering the cooling device from t_0 to t_2 (by *conduction*) and in evaporating the moisture absorbed by the air in passing through the apparatus (by *evaporation*).

The *heat* H_a required to raise the temperature of dry air from t_0 to t_2, B.T.U. per hour, is

$$H_a = Cw(t_2 - t_0). \tag{100}$$

The *heat* H_s required to superheat the water vapor entering the cooling device from temperature t_0 to t_2, B.T.U. per hour, is

$$H_s = w_0 S(t_2 - t_0). \tag{100a}$$

The *heat* H_c abstracted by *conduction* from the circulating water, B.T.U. per hour, is

$$H_c = H_a + H_s. \tag{100b}$$

The *heat* H_e abstracted from the circulating water by *evaporation*, B.T.U. per hour, is

$$H_e = w_3 r_2. \tag{101}$$

Though the process of evaporation is practically continued through the whole range in the cooling device, we are justified in using the heat of vaporization at the highest temperature, because the liquid was at this temperature entering the cooling device and the vapor is brought back to the temperature when leaving it.

The *total heat* H_c absorbed by the air in passing through the cooling device, B.T.U. per hour, is $H_t = H_c + H_e$. (102)

Neglecting radiation and other minor losses, the heat H_t absorbed by the air must be equal to the heat given up by the circulating water, or

$$H_t = H. \tag{103}$$

Example: Determine the quantity of air passing through the cooling tower per hour and the circulating water lost by evaporation in a power plant operating under the following conditions: Engines indicate 500 H.P. and consume 20 lbs. steam per I.H.P. hour; temperature of the injection water, discharge water and outside air, 90, 122 and 72° F., respectively; barometer 29.5; relative humidity of air entering and leaving tower 70 and 90 per cent respectively; vacuum at condenser 25 inches. Determine also the weight of water evaporated in per cent of that circulated and of the condensed steam.

In the problem,

$p_a = 29.5,$	$t_0 = 72,$	† $h_0 = 0.001224,$
† $p_0 = 0.79,$	* $t_2 = 112,$	† $h_2 = 0.003978,$
† $p_2 = 2.79,$	$t_e = 122,$	$z_0 = 0.70,$
$d = 0.0747,$	$t_l = 90,$	$z_2 = 0.90,$
$C = 0.2375,$	$S = 0.45,$	$r_2 = 1028.9.$

These values are obtained from Steam Tables and from Air Tables (Table 58).

Substitute these values in equations (96) to (103) thus:

$$(96),\quad p_1 = 29.5 - 0.79 \times 0.7$$
$$= 28.95.$$
$$(96a), p_3 = 29.5 - 2.74 \times 0.9$$
$$= 27.03.$$
$$(97),\quad w = \frac{28.95}{29.92} \times 0.0747\ V_0.$$
$$= 0.0722\ V_0.$$
$$(97a), w = 0.001224 \times 0.7\ V_0$$
$$= 0.000857\ V_0.$$

* By assumption, t_2 being 10 to 20 degrees lower than t_e in average practice when the range $t_e - t_0$ is greater than 30 degrees.

† Marks and Davis: the values in Table 58 are Regnault's.

$$(98),\ V_2 = \frac{28.95}{27.03} \cdot \frac{460 + 112}{460 + \ \ 72} V_0$$
$$= 1.152;$$

that is, each cu. ft. of dry air entering the cooling-tower is increased in volume to 1.152 cu. ft. as it leaves.

$$(98a),\ w_2 = 0.003978 \times 0.9 \times 1.152\ V_0$$
$$= 0.004125\ V_0.$$
$$(98b),\ w_3 = 0.004125\ V_0 - 0.000857\ V_0$$
$$= 0.003268\ V_0.$$

The total heat to be abstracted from the steam (see equation (84), page 347) is

$$H = 500 \times 20\,(1120.1 - 122 + 32)$$
$$= 10{,}300{,}000 \text{ B.T.U. per hour.}$$
$$(99),\ \text{But } W\,(122 - 90) = 10{,}300{,}000,$$

from which $W = 322{,}000$ pounds per hour.

$$(100),\ H_a = 0.2375 \times 0.0722\ V_0\,(112 - 72)$$
$$= 0.6865\ V_0.$$
$$(100a),\ H_s = 0.000857\ V_0 \times 0.45\,(112 - 72)$$
$$= 0.001543\ V_0$$
$$(100b),\ H_c = 0.6865\ V_0 + 0.001543\ V_0$$
$$= 0.688\ V_0.$$
$$(101),\ H_e = 0.003268\ V_0 \times 1028.9$$
$$= 3.365\ V_0.$$
$$\frac{H_e}{H_c} = \frac{3.365\ V_0}{0.688\ V_0} = 4.89;$$

that is, the *air removes* 4.89 *times more heat by evaporation than by conduction* under the given conditions.

$$(102),\ H_t = 0.688\ V_0 + 3.365\ V_0$$
$$= 4.053\ V_0.$$
$$\frac{H_s}{H_t} = \frac{0.001543\ V_0}{4.053\ V_0} = .00038;$$

that is, the heat required to superheat the moisture carried into the tower by the air is approximately $\frac{4}{100}$ of 1 per cent of the total; hence an error as great as 20% in the mean specific heat of the vapor is negligible.

$$(103),\ 4.053\ V_0 = 10{,}300{,}000,$$

from which

$V_0 = 2{,}543{,}000$ cu. ft. of air per hour necessary to effect the required cooling

$= 42{,}300$ cu. ft. per minute.

From (98b)

$$w_3 = 0.003268\ V_0.$$

Substitute

$$V_0 = 2{,}543{,}000 \text{ in above equation.}$$

$$w_3 = 0.003268 \times 2{,}543{,}000$$
$$= 8320 \text{ lbs., or the weight of circulating water carried away per hour.}$$

$$\frac{w_3}{W} = \frac{8320}{322{,}000} = .0258;$$

that is, 2.58 *per cent of the circulating water is carried away by the air in effecting the necessary cooling.*

$$\frac{w}{20 \times 500} = \frac{8320}{10{,}000} = .832;$$

that is, the equivalent of 83.2 per cent of the steam used by the engines is evaporated in the cooling tower, or the *make-up water is more than supplied by the condensed steam.*

Example: Evaporation Surface Condenser.— How many cubic feet of air and how many pounds of water spray must be forced through an evaporative surface condenser of the fan type in order to condense 1000 pounds of steam per hour and maintain a vacuum of 25 inches, barometer 29? (Atmospheric air 80° F., relative humidity 70%.) The air and vapor issue from the discharge pipe under pressure of four inches of water, temperature 120° F., relative humidity 98%.

The absolute pressure in the condenser is 29.0 − 25.0 = 4 inches of mercury.

The total heat to be withdrawn in order to cool and condense 1000 lbs. of steam per hour at absolute pressure of 4 inches to 120° F. is

$$1000\,[1114.8 - (120 + 32)] = 1{,}026{,}000 \text{ B.T.U.}$$

Neglecting radiation and leakage losses, this is the heat to be abstracted per hour by the air and water spray.

The pressure of the dry air in the mixture entering the condenser is, equation (96),

$$p_1 = 29.0 - 0.7 \times 1.029.$$
$$= 28.28.$$

The pressure of dry air in the mixture leaving the condenser is, equation (96a),

$$p_3 = (29.0 + 0.294) - 0.98 \times 3.438$$
$$= 25.925$$

(0.294 is the value in inches of mercury of four inches of water-fan pressure).

Let V_0 = volume of atmospheric air *entering* the condenser. The volume leaving the condenser will be, equation (98),

$$V_0 = \frac{28.280}{25.925} \cdot \frac{460 + 120}{460 + 80} = 1.172\ V_0.$$

The weight of vapor in the condenser discharge is, equation (98a),

$$\begin{aligned} w_2 &= 1.172\ V_0 \times 0.004888 \times 0.98 \\ &= 0.005615\ V_0 \text{ lbs.} \end{aligned}$$

The weight of vapor in the mixture entering the condenser is, equation (97a),

$$\begin{aligned} w_0 &= 0.00157 \times 0.7\ V_0 \\ &= 0.001099\ V_0 \text{ lbs.} \end{aligned}$$

The amount evaporated therefore is

$$\begin{aligned} w_3 &= 0.005615\ V_0 - 0.001099\ V_0 \\ &= 0.004516\ V_0 \text{ lbs.} \end{aligned}$$

The *weight* of *dry air* entering the condenser is, equation (97),

$$\begin{aligned} w &= \frac{28.28}{29.921} 0.07362\ V_0. \\ &= 0.06958\ V_0 \text{ lbs.} \end{aligned}$$

The *heat* absorbed by the *dry air* in being heated from 80° to 120° F. is, equation (100),

$$\begin{aligned} H &= Cw\,(t_2 - t_0) \\ &= 0.02375 \times 0.06958\ V_0\,(120 - 80) \\ &= 0.658\ V_0 \text{ B.T.U.} \end{aligned}$$

Heat required to superheat w_0 lbs. of vapor from 80° to 120° F. is, equation (100a),

$$\begin{aligned} H_s &= 0.001099\ V_0 \times 0.46\,(120 - 80) \\ &= 0.02022\ V_0 \text{ B.T.U.} \end{aligned}$$

Heat absorbed by the evaporation of w_3 lbs. of water is, equation (101),

$$\begin{aligned} H_e &= 0.004516\ V_0 \times 1046.7 \\ &= 4.720\ V_0 \text{ B.T.U.} \end{aligned}$$

(Here the latent heat is taken at the lower temperature, it being the original temperature of the liquid.)

Total heat absorbed by the entering air and spray is

$$\begin{aligned} H_t &= 0.658\ V_0 + 4.720\ V_0 + 0.020\ V_0 \\ &= 5.398\ V_0. \end{aligned}$$

But this represents also the heat given up by the steam, or

$$5.398\ V_0 = 1{,}026{,}000.$$

From which V_0 = 190,500 cu. ft. of atmospheric air necessary to condense and cool 1000 pounds of steam under the given conditions.

The water spray to be injected per hour is

$$0.004516\ V_0 = 0.004516 \times 190{,}500 = 860 \text{ pounds.}$$

236a. Hygrometry. — The degree of saturation, or relative humidity, is ordinarily determined from the difference in reading of a wet and a dry bulb thermometer, thus: If the air is saturated with aqueous vapor no evaporation takes place from the wet bulb and the two thermometers give identical readings; but if it is unsaturated, evaporation occurs. The wet-bulb thermometer is thus cooled and its readings are lower than those of the dry bulb. The difference in reading is a function of the relative humidity, and the latter may be calculated from the following modification of *Apjohns'* formula:

If the thermometer reads above 32° F.

$$h = \left(p_w - \frac{dP}{2640}\right)\frac{100}{P_t}. \tag{103a}$$

If it reads below 32° F.

$$h = \left(P - \frac{dP}{2880}\right)\frac{100}{P}, \tag{103b}$$

in which

h = relative humidity, per cent.

d = difference in reading of the wet and dry thermometers, degrees F.

P = barometric pressure, inches of mercury.

P_w = maximum tension of aqueous vapor corresponding to the temperature of the wet thermometer, inches of mercury. (This may be taken directly from the Steam Tables.)

P_t = maximum tension of aqueous vapor corresponding to the temperature of the dry thermometer, inches of mercury.

Example: Determine the relative humidity when the dry bulb reads 70° F., wet bulb 60° F., barometer 28.0.

From the Steam Tables we find

$$P_w = 0.522; \quad P_t = 0.739.$$

Whence

$$h = \left(0.522 - \frac{10 \times 28}{2640}\right)\frac{100}{0.739} = 56.5 \text{ per cent.}$$

Tables giving the relative humidity in terms of the temperature difference are published in most engineering handbooks and the above calculations are unnecessary. These tables, however, are based on a *fixed* barometer pressure, whereas the formula takes the *actual* pressure into consideration.

237. Test of Cooling Tower (Wheeler Condenser Company). — The following gives the results of a test made on the cooling-tower plant of the A. F. Brown Company at Elizabethport, N. J. The tower is working in connection with a Wheeler surface condenser of 280 square feet of cooling surface, mounted over a 10, 12 × 12 combined air and circulating pump.

Observations made on June 24, 1904.

Temperature of air....................81 degrees
Hygrometer....................69 degrees
Temperature of air at top of tower....................89 degrees
Temperature of water in troughs....................105 degrees
Temperature of water in tank....................83 degrees
Revolutions of fan, 239 r.p.m., air pressure....................$\frac{3}{4}$ inch water
Velocity of air out of tower....................822 feet per minute
Gallons of water passing over mats....................385 per minute
Vacuum....................26 inches
Temperature of air-pump discharge....................87 degrees

Observations made June 28, 1904, 9 A.M

Temperature of air....................76 degrees
Hygrometer....................59 degrees
Temperature of air at top of tower....................81 degrees
Temperature of water in troughs....................96 degrees
Temperature of water in tank....................78 degrees
Revolutions of fan, 232 r.p.m., air pressure....................$\frac{3}{8}$ inch water
Velocity of air out of tower....................680 feet per minute
Gallons of water passing over mats....................406 per minute
Vacuum....................25.5 inches
Temperature of air-pump discharge....................90 degrees

Observations made June 28, 1904, 3 P.M.

Temperature of air....................74 degrees
Hygrometer....................57 degrees
Temperature of air at top of tower....................83 degrees
Temperature of water in troughs....................99 degrees
Temperature of water in tank....................80 degrees
Revolutions of fan, 237 r.p.m , air pressure....................$\frac{5}{16}$ inch water
Velocity of air out of tower....................769 feet per minute
Gallons of water passing over mats....................470 per minute
Vacuum....................25.5 inches
Temperature of air-pump discharge....................92 degrees

Observations made June 29, 1904.

Temperature of air....................78 degrees
Hygrometer....................71 degrees
Temperature of air at top of tower....................86 degrees
Temperature of water in troughs....................108 degrees
Temperature of water in tank....................82 degrees
Revolutions of fan, 241 r.p.m., air pressure....................$\frac{3}{8}$ inch
Velocity of air out of tower....................772 feet per minute
Gallons of water passing over mats....................430 per minute
Vacuum....................25.5 inches
Temperature of air-pump discharge....................93 degrees

Specifications for condensers — See paragraph 414.

RESULTS OF TEST OF NATURAL-DRAFT TOWER, DETROIT.

COMPLETE FIVE-FIFTHS SURFACE INSTALLED.

Proc. A.S.M.E., Mid-Nov., 1909, p. 1205.

Engines:	Two 400-i.h.p. 300-kw. MacIntosh & Seymour tandem-compound engines, overhung generators.
Condensers:	Worthington surface (admiralty type) 1600-sq. ft. reciprocating wet-air pump and circulating pump
Tower:	Wood-mat construction, 24,500 sq. ft evaporating surface, exclusive of shell
Test:	March 15 to 16, 1901, 4 p.m. to 4 p m , 24 hr.

		A.M.		P.M.	AVERAGE.
Weather:	Barometer (abs.), min.......	30.22	30.07;	30.14	30.27
	Temperature air, deg........	18.5	25;	30	25
	Relative humidity, per cent .	76	82;	58	72

Load:	600 kw. max. to 50 kw. min. Average..............	244.9 kw.
	Engine efficiency = 92.5 = 875 i.h.p. max. Average ..	354.8 i.h.p.
Steam:	Weight of condensed steam per hr., lb................	5910.6
	Temperature exhaust steam, deg. F.................	134.38
	Temperature condensed steam, deg. F................	108.78
	Weight of steam per hour, max. load, lb.............	13,500
	Vacuum (abs.) 25 to 19, average about...............	22
	Vacuum corresponding to temperature exhaust steam...	25
	Vacuum possible with good condenser (10 deg. difference)	28
Water:	Circulated per hr., lb..............................	293,536
	Temperature hot well, average, deg. F................	87.50
	Temperature cold well, average, deg. F...............	71.27
	Vaporization loss per hr., lb........................	5970
Results:	Condenser surface per kw., sq. ft.....................	2.66
	Steam per kw. hr., lb...............................	24.3
	Steam per i.h.p. hr., lb.............................	16.66
	Circulating water per lb. of steam, lb................	49.6
	Steam per sq. ft. condenser surface per hr., lb..........	3.7
	Circulating water per sq. ft. tower surface, lb...........	12
	Difference in temperature between exhaust steam and discharge, deg. F...............................	47
Cooling:	Max. 20 deg., min. 3 deg.-5 deg. Average.............	16.23
	Heat dissipated per hr., B.T.U.......................	4,769,000
	Heat per sq. ft. tower surface, B.T.U................	195
	Heat per sq. ft. per 1000 lb. water, B.T.U..............	0.665
Evaporation:	Circulating water, per cent	2.03
	Engine steam, per cent.............................	101
Tower:	Surface per kw. (average load 245 kw.), sq. ft..........	100
	Surface per kw. (max. load 600 kw.), sq. ft.............	40.8
	Surface per 1000 lb. steam max. load, sq. ft............	1820
	Surface per 1000 lb. steam average load, sq. ft..........	4140
	Surface per 1000 lb. circulating water per deg. max. cooling, sq. ft..................................	4.17

Time.	Temperature, Deg. Fahr.					Quantities.				
	Air.	Hot Well.*	Cold Well.	Water Cooling.	Total Heat Head.†	Tower Water, Lb. per Hr.	Heat Dissipated, B.t.u. Lb. per Hr.	Heat per Sq. Ft. Cooling Surface, B.t.u. per Hr.	Circulating Water per Sq. Ft., Lb. per Hr.	Load, Kw.
1	2	3	4	5	6	7	8	9	10	11
12 noon	34	102	89	13	68	375,000	4,880,000	332	25	270
1.30	35	106.5	90	16.5	71.5	‡{375,000 370,200	6,108,000	415	24.8	{315 290
2.30	35	106.5	87.5	19	71.5	375,000	7,120,000	484	25	315
3.30	35	113	88.5	24.5	78	375,000	9,000,000	613	25	350
4.30	32.5	100	84	16	67.5	399,000	6,384,000	434	26.6	365
5.00	28.5	103.5	88	15.5	75	445,500	6,900,000	470	29.7	485
6.00	26	125	94	31	99	417,000	12,930,000	880	27.8	655
7.00	24	121	94	27	97	427,000	11,532,000	785	27.4	570
8.00	24	123	94.5	28.5	99	427,000	12,174,000	827	27.4	600

* Assuming a more efficient condenser, say 10 deg. difference, the probable vacuum would be 20 deg. to 27.5 deg. This condenser actually operated at 40 deg. to 50 deg. difference.

† Total heat head = air heating + lost head. ‡ Difference due to rapid change in load.

CHAPTER XII.

FEED-WATER PURIFIERS AND HEATERS.

238. General. — All natural waters contain more or less foreign matter either in suspension or solution. Waters containing carbonates and sulphates of magnesia and lime, soluble salts of silica, iron, and alumina, and suspended matter, tend to form scale in the boiler and reduce its steam-generating capacity and economy. The loss due to this cause is often overestimated but is of secondary importance to the danger due to retarded heat transmission which overheats and weakens the plates and tubes.

Table 59 gives the results of a number of tests made on locomotive boiler tubes with different thicknesses and characters of scale. The diversity of the results indicates the futility of basing the decrease in conductivity on the thickness of the scale. For example, test No. 1 shows a decrease in conductivity of 9.1 per cent for a scale .02 inch thick, while No. 16 shows a decrease of only 6.75 per cent for a scale over 6.5 times as thick. The scale in each case was even, hard, and dense. Again, No. 8 with a very soft scale .042 inch thick gives a decrease in conductivity of 9.54 per cent, whereas No. 14, also very soft but twice as thick, gives a decrease of only 4.95 per cent. No doubt the heat transmission is a function of the chemical as well as the physical properties, but further experiments are necessary before any specific conclusion can be drawn.

Waters containing acids, organic matter, and magnesium chloride and sulphate tend to corrode the boiler, and those containing sodium carbonate, organic matter, and alkalies induce priming. Even distilled water, as obtained from a surface condenser, is a solvent of iron to a certain extent and causes corrosion and pitting. Table 60 gives some idea of the character and extent of impurities in water from various localities, with an analysis of the scale produced by the water and the trouble in the boiler arising from its use.

It is impossible to judge the quality of feed water merely by the grains of solids per gallon, since a large amount of soluble salt such as sodium chloride will not be as deleterious as a very small amount of calcium sulphate.

TABLE 59.

INFLUENCE OF SCALE ON HEAT TRANSMISSION.

(Locomotive Boiler Tubes.)

No.	Thickness of Scale, Inches.	Character of Scale.	Decrease in Conductivity due to Scale. Per cent.
1	.02	Hard, dense	9.1
2	.02	Hard	2.02
3	.033	Soft	4.3
4	.033	Very hard	3.5
5	.038	Medium	4.03
6	.04	Soft, porous	6.82
7	.04	Hard, dense	3.07
8	.042	Very soft	9.54
9	.047	Hard	2.75
10	.065	Medium	2.39
11	.07	Soft	2.38
12	.07	Hard	4.43
13	.085	Soft, porous	19.0
14	.089	Very soft	4.95
15	.11	Hard, porous	16.73
16	.13	Hard, dense	6.75

From tests conducted at the University of Illinois, *Railroad Gazette*, Jan. 27, 1899, June 14, 1901. See also *Engineering Record*, Jan. 14, 1905, p. 53; *Power*, February, 1903, p. 70; *Street Railway Review*, July 15, 1901, p. 415.

The following is a rough rating according to the number of grains of incrusting solids per United States gallon:

Less than
- 8 grains .... very good.
- 12 to 15 grains .... good.
- 15 to 20 grains .... fair.
- 20 to 30 grains .... bad.
- Over 30 grains .... very bad.

This applies to calcium carbonate, magnesium carbonate, and magnesium chloride. For water containing sulphate of calcium and magnesium, divide the first column by 4 for the same rating.

On account of the great variety of possible impurities the proper treatment to be adopted can be determined only by chemical analysis of the feed water in each case.

Table 61, compiled by the Hartford Steam Boiler Inspection and Insurance Company, shows the number of boilers inspected by that company during the year 1907 and the number found defective from various causes.

TABLE 60.

WATER AND BOILER SCALE ANALYSES.

Water Analysis. Grains Per U.S. Gallon.	Lake Michigan, Chicago.	Well 115 Feet, St. Bernard, Ohio	San Francisco, Cal.	Schuylkill River, Philadelphia.	Camaguey, Cuba.	Park City, Utah.	Toledo, Ohio.	Kewanee, Ill.	Arkansas River, Florence, Colo.	Surface Water, Auburn Park, Ill.
	1	2	3	4	5	6	7	8	9	10
Silica	0.438	0.677	0.759	0.338	2.873	1.354	0.759	0.373	0.630	0.508
Oxide of iron and aluminum	0.099	0.116	0.116	0.093	0.140	0.350	0.163	0.081	0.075	0.175
Carbonate of lime	3.731	2.271	4.207	0.068	10.270	1.476	5.519	1.721	2.158	2.382
Sulphate of lime	0.962	4.083	0.680	2.257	3.220	1.360	3.950	1.360	18.540	3.154
Carbonate of magnesia	2.092	4.424	2.866	0.884	4.900	0.318	2.592	2.212	4.848	2.875
Sodium and potassium sulphates	Trace	Trace	1.681	Trace	Trace	0.867	Trace	12.928	11.319	Trace
Sodium and potassium chlorides	0.670	0.990	2.970	0.990	5.708	1.980	2.740	26.070	2.028	1.650
Organic matter	0.066	0.584	......	0.700	33.000	2.569	1.052	0.584	0.701	0.584
Total mineral matter	8.058	12.614	13.665	4.672	32.288	7.826	15.885	45.318	40.062	11.096
Chloride of magnesia	......	......	......	......	5.6	......	......	......	......	......

SCALE ANALYSIS — PER CENT.

	1	2	3	4	5	6	7	8	9	10
Character of sample	Hard, brittle.	Medium hardness	Hard, brittle.	Hard, impervious	Very hard.	Very hard.	Hard.	Soft, brittle.	Hard, crystalline.	Medium hardness
Silica	20.60	8.44	11.18	12.30	24.42	19.00	4.96	2.52	6.20	5.7
Oxide of iron and aluminum	10.30	1.30	10.44	6.18	1.02	6.26	11.80	4.92	2.36	2.04
Carbonate of lime	33.86	37.22	40.96	21.26	29.10	29.02	3.74	18.18	18.78	29.86
Sulphate of lime	None	33.82	Trace	34.62	0.96	5.48	55.38	54.76	59.84	39.64
Carbonate of magnesia	6.04	Trace	22.60	Trace	Trace	Trace	8.19	Trace	0.84	Trace
Magnesia (MgO)	15.48	12.01	......	8.20	25.94	1.45	6.86	9.08	4.75	13.8
Moisture and organic matter	12.89	6.22	13.56	11.70	16.66	13.69	8.69	7.40	5.73	7.04
Oil	Trace	......	......	0.27	......	......	......	2.92	......	......
Loss and undetermined	0.83	0.99	1.24	0.23	1.90	1.55	0.38	0.22	1.50	1.32
Lime (CaO)	......	......	......	5.24	......	23.55	......	......	......	......

1. This water will cause the deposit of a moderate amount of scale which will be hard and persistent.

2. This water will cause a large amount of scale to deposit.

3. This water will cause a moderate amount of scale with a decided tendency to galvanic action on account of the large proportion of sodium and potassium salts present.

4. This water will cause the formation of a moderate amount of very hard scale.

5. This water will cause the deposition of a moderate amount of hard

TABLE 61.

SUMMARY OF INSPECTORS' REPORTS FOR THE YEAR 1907.

(Hartford Steam Boiler Inspection and Insurance Company.)

Nature of Defects.	Whole Number.	Dangerous.
Cases of deposit of sediment	18,917	1,315
Cases of incrustation and scale	38,427	1,333
Cases of internal grooving	3,010	258
Cases of internal corrosion	12,802	528
Cases of external corrosion	10,230	768
Defective braces and stays	2,219	578
Settings defective	6,363	699
Furnaces out of shape	7,564	396
Fractured plates	3,551	568
Burned plates	4,878	499
Laminated plates	898	92
Cases of defective riveting	3,582	823
Defective heads	1,764	238
Leakage around tubes	11,357	1,599
Cases of defective tubes	8,266	3,054
Tubes too light	1,947	563
Leakage at joints	5,557	430
Water gauges defective	3,008	707
Blow-offs defective	4,216	1,250
Cases of deficiency of water	413	156
Safety valves overloaded	1,231	415
Safety valves defective	1,211	407
Pressure gauges defective	7,651	465
Without pressure gauges	194	194
Unclassified defects	27	10
Totals	159,283	17,345

The neutralization or elimination of the impurities may be effected by one of the following methods:

1. Chemically.
   - Boiler compounds.
   - Purifying plants.
2. Mechanically.
   - Filters.
   - Blow-off.
   - Tube cleaners.
3. Thermally.
   - Feed-water heater.
   - Distillation.

The following chart ("Boiler Waters," W. W. Christie) outlines some of the troubles arising from feed water, their cause and means for preventing them.

Trouble.	Cause.	Remedy or Palliation.
Incrustation.	Sediment, mud, clay, etc.	Filtration. Blowing off.
	Readily soluble salts	Blowing off.
	Bicarbonate of magnesia, lime, iron	Heating feed and precipitate. Caustic soda. Lime. Magnesia.
	Organic matter	See below.
	Sulphate of lime	Sodium carbonate. Barium chloride.
Corrosion.	Organic matter	Precipitate with alum / Precipitate with ferric chloride } and filter
	Grease	Slaked lime / Carbonate of soda } and filter
	Chloride or sulphate of magnesium	Carbonate of soda.
	Sugar / Acid	Alkali.
	Dissolved carbonic acid and oxygen	Slaked lime. Caustic soda. Heating.
	Electrolytic action	Zinc plates.
Priming.	Sewage	Precipitation with alum or ferric chloride and filter.
	Alkalies	Heating feed and precipitate.
	Carbonate of soda in large quantities	Barium chloride.

Analysis of Water for Softening by Chemical Processes: Jour. Soc. Chem. Ind., May, 1899, p. 520. *Volumetric Determination of Calcium and Magnesium in Water:* Jour. Soc. Chem. Ind., May 31, 1901, p. 507. *Simple Tests for Boiler Water:* Soc. Engng., May, 1904, p. 238. *Qualitative Tests of Feed Water:* Power, Dec., 1904, p. 756. *New Testing Apparatus for Boiler Feed Water:* West. Elecn., Aug. 1, 1903, p. 85. *A Simple Method of Calculating Water Analyses and Amounts of Substances to be Added for Preventing Scale and Corrosion:* Jour. Frank. Inst., Vol. CLIX, p. 217. *Description of Dearborn Drug and Chemical Co.'s Laboratories for Analyses of Boiler Feed Water:* St. Ry. Review, Sept. 15, 1901.

Boiler Corrosion: Power, Jan., 1906, Oct., 1905, p. 591, Dec., 1900; Eng. Rev., Oct., 1904, p. 12; Eng. Mag., Dec., 1905, p. 425, Oct., 1900, p. 118; Engng., Oct. 10, 1902, p. 482; Engr. U.S., Jan. 1, 1907, p. 103, May 1, 1902, p. 388; Engr., Lond., Aug. 5, 1904, p. 131, July 29, 1904, p. 115, Dec. 4, 1896, p. 574; Elec. Age, Dec., 1905, p. 456; Am. Elecn., Aug., 1905, p. 436, April, 1902, p. 184; Jour. Am. Soc. Nav. Engrs., May, 1902; Mines and Min., Sept., 1903; Stahl u. Eisen, Jan. 15, 1904.

Boiler Incrustation: Am. Elecn., April, 1904, p. 206, Dec., 1901, p. 576, May, 1901, p. 220, Oct., 1898, p. 473; Cassier's Mag., July, 1903, p. 273; Chem. News,

Oct. 18, 1901, p. 191; Engr., Lond., Jan. 21, 1898, p. 52; Engr. U.S., April 16, 1906, March 15, 1904, p. 202, May 15, 1904, p. 354, Sept. 1, 1904, p. 608; Ice and Refrig., Nov., 1905, p. 173. *Chemistry of Scale:* Jour. Frank. Inst., Aug., 1901, p. 113, Aug. 1891, p. 145. *Boiler Scale:* Power, Dec., 1905, p. 779; St. Ry. Review, April 2, 1904, p. 545. *Scale Prevention:* Am. Elecn., Dec., 1901, p. 578; Am. Engr., May, 1900, p. 138; Eng. Mag., 1897, 12–959, 13–74, 232, 419; Elec. Engr., Lond., July 20, 1900, p. 91; Engrs. Gaz., July, 1902.

Foaming. — *Foaming Water and Scaling Water for Locomotive Boilers:* Eng. News, July 21, 1904, p. 71, Sept. 1, 1904, p. 198; *Foaming and Priming:* Soc. Nav. Arch. and Marine Engrs., 1902; R.R. Gazette, Oct. 12, 1900; Christie, Boiler Waters, Chap. V; Stromeyer, Steam Boilers, pp. 67–83; Rowan, Modern Steam Boilers, pp. 321–354.

239. Chemical Purification. — Chemical treatment of boiler feed water has been remarkably developed during recent years and a number of manufacturing concerns make this their sole business. The two most common systems of chemical treatment involve (1) boiler compounds and (2) purifying plants. In the former the necessary chemical action takes place inside the boiler and in the latter the water is purified before it enters the boiler. In either case the usual procedure is to submit for analysis a sample of the feed water and the resulting scale to a competent chemist who will specify the character and quantity of chemicals necessary to bring about the desired result.

240. Boiler Compound. — The object of treatment with boiler compounds is to neutralize the evil effects of the impurities in the feed water or to change them into others which are less objectionable and which are easily removed. When properly compounded and introduced into the boiler such preparations are of great benefit and practically overcome the deleterious effects, but when improperly used they may produce even greater troubles than the impurities which they are expected to eliminate.

Boiler compounds may be divided into three classes:

1. Those converting the scale-forming elements into new substances which will not form a hard, resisting scale and which are readily removed by skimming, blowing off, or by tube cleaners. For example, feed water containing sulphates of lime and magnesia will form a dense, tenacious scale. If carbonate of soda be added in correct amount, the sulphates are converted into insoluble carbonates which are precipitated and form scale varying from a more or less porous, friable crust to a soft " mush " or mud. The resulting sulphate of soda remains in solution and does not form scale unless allowed to concentrate, and this is prevented by blowing off. An excess of soda

is apt to cause foaming and at high temperatures is liable to attack the inside of gauge glasses. Bisodium and trisodium phosphate, sodium tannate, fluoride of sodium, sugar, etc., have all proved satisfactory, but as each case requires special treatment no detailed discussion is possible within the scope of this work and the reader is referred to the accompanying bibliography.

2. Those enveloping the newly precipitated scale-forming crystals with a surface which prevents them from cementing together. The ingredients used to bring about this result are starches, woody fibers, dextrine, slippery elm, and the like.

3. Those preventing the formation of hard scale by a solvent or "rotting" action, as kerosene and petroleum oils.

Boiler Compounds. — *Use of Compounds:* Eng. News, July 27, 1905, p. 112; Am. Mach., Dec. 7, 1899, p. 115, Oct. 26, 1899, p. 1014; Power, Aug., 1903; Eng. and Min. Jour., Aug. 12, 1905, p. 253.

241. Use of Kerosene and Petroleum Oils in Boiler Feed Water. — Kerosene oil and other refined petroleum oils are sometimes used with good effect in boilers to prevent scale from adhering. These oils are said to change the deposit of lime from a hard scale to a friable material which may be easily removed. They are ordinarily fed to the boiler with the feed water, drop by drop, through a sight feed apparatus similar to a cylinder oil lubricator. From extended experiments made on a 100-horse-power tubular boiler fed with water containing 6.5 grains of solid matter per gallon it was found that one quart of kerosene per day was sufficient to keep the boiler entirely free from scale. Prior to the introduction of the oil the water had a corrosive action upon some of the fittings attached to the boiler, but after the oil had been used for a few months it was found that the corrosive action had ceased. In another case 40 gallons of kerosene were used in 24 hours in a steamer of about 3000 horse power. These boilers showed no incrustation but considerable corrosion. Evidently oil does not have the same effect or give the desired results in all cases. Kerosene used in moderate quantities will not cause foaming. Crude oil should never be used, as the heavy residue causes the formation of a tough, impervious scale productive of bagged sheets and collapsed flues.

Use of Kerosene in Boilers: Engr. U.S., Sept. 15, 1905, p. 634; Eng. News, May 24, 1890, p. 497; Power, Aug., 1895, p. 13, May, 1896, p. 16; Trans. A.S.M.E., 9-247, 11-937; Locomotive, July, 1890, p. 97.

242. Use of Zinc in Boilers. — Zinc is often introduced into boilers to prevent corrosion. The theory is that a feeble but continuous cur-

rent of hydrogen is generated over the whole extent of the iron by electrolytic action. The bubbles of hydrogen formed isolate the metallic surface from scale-forming substances. If there is but a little of the scale-forming element it is precipitated and reduced to mud; if there is considerable, coherent scale is produced which takes the form of the iron surface but does not adhere to it, being prevented from doing so by the intervening bubbles of hydrogen. Zinc is ordinarily suspended in the water space of the boiler in the shape of blocks, slabs, or as shavings in a perforated vessel. Electrical connection between the metallic surfaces is essential. Rolled zinc slabs 12 x 6 x ½ inches have found much favor in marine practice. Generally speaking one square inch of zinc surface is sufficient for every 50 pounds of water in the boiler, though the quantity placed in the boiler should vary with the hardness. The British Admiralty recommends the renewing of the zinc slabs whenever the decay has penetrated to a depth of ¼ inch below the surface. Zinc does not prevent corrosion or scale formation in all cases and may even aggravate the trouble.

Use of Zinc in Boilers: Am. Elecn., Dec., 1901, p. 572; Kent, Steam Boiler Economy, p. 318; Christie, Boiler Waters, p. 137; Stromeyer, Marine Boiler Management and Construction, p. 81.

243. Methods of Introducing Compounds. — Boiler compounds may be introduced into the boiler continuously or intermittently. Small quantities introduced continuously or at short intervals are more effective than large quantities at long intervals. Continuous feeding is ordinarily brought about by connecting the suction side of the feed pump with a reservoir containing the compound in solution, arranged similarly to an ordinary cylinder oil lubricator. In large plants an independent pump is often used to force the solution into the feed line. Intermittent feeding is brought about by temporarily connecting the suction of the feed pump with the reservoir containing the compound. The use of boiler compounds does not necessarily prevent scale from forming in time, though it will reduce the evil to a minimum. In some instances where compounds are used it is found necessary to run a tube cleaner through the tubes at certain intervals, in others such a course has not been found necessary.

244. Weight of Compound Required. — The weight of compound introduced depends upon the nature of the reagents used and the character of the feed water, and ranges from a few ounces to several pounds per 100 gallons of feed water. For example, water containing 4 grains of calcium sulphate and 6 grains of magnesium sulphate per gallon will require 3.57 pounds of carbonate of soda per 1000 gallons

of water for the reduction of the sulphates. The chemical reaction and analysis is as follows:

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4$$
$$MgSO_4 + Na_2CO_3 = MgCO_3 + Na_2SO_4$$

If x = grains of Na_2CO_3 necessary for the calcium,

$$CaSO_4 : Na_2CO_3 + 10H_2O = 4 : x.$$
$$40 + 32 + 4 \times 16 : 2 \times 23 + 12 + 3 \times 16 + 10\,(2 + 16) = 4 : x.$$
$$x = 8.41 \text{ grains.}$$

If y = grains of Na_2CO_3 necessary for the magnesium,

$$MgSO_4 : Na_2CO_3 + 10H_2O = 6 : y.$$
$$24 + 32 + 4 \times 16 : 2 \times 23 + 12 + 3 \times 16 + 10\,(2 + 16) = 6 : y.$$
$$y = 14.3.$$

The total weight of carbonate of soda per 1000 gallons is therefore

$$1000\,(14.3 + 8.41) = 22{,}710 \text{ grains.}$$
$$= 3.24 \text{ pounds.}$$

This amount would effect the desired result if the chemical reaction is permitted to take place for some time, otherwise an excess of reagent is necessary.

245. Mechanical Purification. — Waters containing sand, mud, organic matter, and in fact all matter which is not in solution or in chemical combination with the water may be purified by mechanical filtration. Mud and sand may be eliminated by simply permitting the water to stand for some time in settling tanks. Suspended matter which will not gravitate to the bottom may be removed by filtering the water through coke, cloth, excelsior, or the like. Filters should be in duplicate for continuity of operation.

Vegetable and other organic impurities commonly float on the surface of the water when the boiler is making steam, and may be blown out through a "surface blow-out." (See paragraph 82.)

Precipitated matter may be ejected from the boiler by frequent blowing off before it has time to adhere and bake to a crust. This procedure is particularly essential when boiler compounds are used.

For description and use of mechanically operated tube cleaner see paragraph 86.

246. Thermal Purification. — (See also Live Steam Purifiers, paragraph 263.) The carbonates of lime and magnesia are held in solution

in fresh water by an excess of carbon dioxide and are completely precipitated by boiling. At ordinary temperatures carbonate of lime is soluble in approximately 20,000 times its volume of water, at 212 degrees F. it is slightly soluble, and at 290 degrees it is insoluble. Sulphate of lime is much more soluble in cold than in hot water, and is completely precipitated at 290 degrees. (*Revue de Mécanique*, November, 1901, pp. 508, 743.) Thus it will be seen that a feed heater may be relied upon to remove part or all of the lime, depending upon the temperature to which the water is raised and the time in which the precipitation is permitted to take place.

Influence of Temperature and Concentration on the Saline Constituents of Boiler Water: Jour. Soc. Chem. Ind., Oct. 31, 1900, p. 885. *Solubility of Sulphate of Lime:* Rev. de Mécanique, Jan., 1901, p. 5, Nov., 1901, p. 508.

247. Purifying Plants. — The function of a purifying plant is the elimination of all impurities from the feed water before it enters the boiler. In the Scaife system for water purification feed water first enters the heater, where it attains a temperature of from 200 to 210 degrees F. As a portion of the free CO_2 is driven off by the heat the carbonates of lime and magnesia are precipitated and are deposited in removable pans inside the heater. On its way the heated water is forced by the boiler feed pump into a large precipitating tank, where the necessary chemicals are introduced by means of two small pumps. These pumps take the solution of chemicals from the solution tanks which hold a sufficient quantity to operate the plant from eight to twelve hours. The precipitating tank is so constructed as to cause intimate and thorough mixing of the chemicals with the water. Thus the acids are neutralized, and the scale-forming substances are precipitated by being changed to insoluble substances which sink to the bottom of the precipitating tank whence they are readily removed. Some of the lighter substances remaining in suspension are carried along with the water as it passes into the filters, which effectively remove all suspended matter. This system is continuous in operation, and purification is accomplished without appreciably retarding the onward flow of feed water. Fig. 237 shows a modification of the system. The chemicals are pumped from the "chemical tank" into the "solution tanks," where the feed water and chemical solution are thoroughly mixed. The treated water is taken from these tanks and pumped into the "precipitating tanks" where a large portion of the scale-forming element is precipitated. From the precipitating tanks the water is forced through a series of filters to the boiler.

Fig. 238 illustrates the We-Fu-Go system of water purification. In

PRECIPITATING

FIG. 237. General Arrangement of Scaife System of Feed-Water Purification.

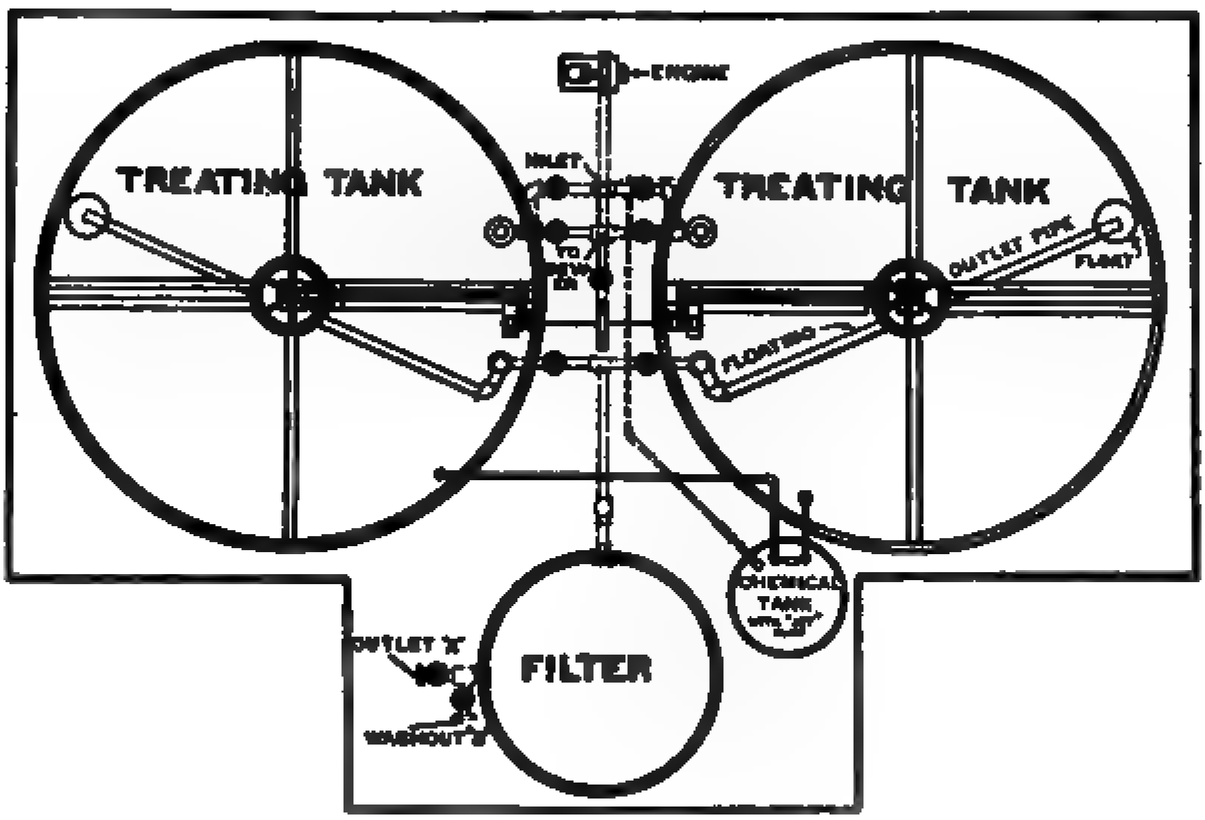


FIG. 238. General Arrangement of We-Fu-Go System of Feed-Water Purification.

this installation the water supply first enters the settling or treating tanks into which the chemicals are fed. A thorough mixture is effected by the use of the two armed paddles located near the bottom of the tanks. From the treating tanks the water flows by gravity into the filters, which remove all remaining impure solid matter which does not settle to the bottom of the treating tank. The pipes conducting the water from the settling tanks to the filter are fitted with a flexible joint and float so that the outlets are near the surface at all times, rising and falling with the water level. From the filters the purified water gravitates into the clear water storage reservoir, from which it is pumped into an open heater and thence to the boiler. This system is intermittent in operation, and in order to provide sufficient time for thorough chemical treatment of large quantities, two or more settling tanks are employed. Both the We-Fu-Go and Scaife systems are modified in a number of ways to meet different conditions.

FIG. 238a. Anderson System for Preventing Corrosion in Condensers.

Fig. 238a shows the general arrangement of the Anderson system for preventing corrosion in condensers and removing oil from condensed steam. The method consists in injecting into the exhaust steam as it passes from the preheater to the condenser a solution containing a coagulant which changes the emulsion of the cylinder oil to a flaky condition so that it may be separated by settling, flotation, or filtering. The air pump delivers the water to the settling tank *F*, whence it is taken to the open gravity filters *G*, *G*, of a superficial area proportional to the amount of water to be passed and containing a filter bed of four feet of crushed quartz. This will run about four days without any marked difference in efficiency, after which time the bed is stirred to a depth of two feet by mechanical agitators and flushed with clean water, by which all impurities are carried to the sewer. The solution is pre-

pared in tank A, in which the water level is preserved by a ball float and into which filtered water is admitted through pipe B, while the substance with which the water is treated is pumped in through the pipe D by a small pump operated from the main engine. The flow to the "rose head" above the condenser is controlled by the valve E, and a meter in this pipe records the amount being fed. The water ordinarily required for "make up" is sufficient to carry in the solution. There is very little loss of water, and the rapid corrosion of the condenser tubes, which has been so great an obstacle to the successful use of surface condensers, is much reduced. The chemicals used perform a twofold duty, viz., to neutralize the water and make it chemically inactive and to coagulate the oily matter contained in the steam so that mechanical filtration is possible. (*Power*, June 1903, p. 304.)

Water-softening plants cost from \$4 to \$5 per horse power for plants of 1000 horse power and less, from \$3 to \$4 for plants of 1000 to 2000 horse power, and as low as \$1.50 for plants of 5000 horse power or more. The depreciation of wooden tanks is as high as 15 per cent a year, while that of steel tanks should not be greater than 5 per cent. Unless wooden tanks are considerably cheaper than steel tanks they are not a good investment. The cost of water purification varies from a fraction of a cent to 2 cents per 1000 gallons, depending upon the size of the plant and the quantity and character of the impurities. (*American Electrician*, March, 1905, p. 125.)

Feed Water Purification: Am. Elecn., March, 1900, p. 145, April, 1900, p. 190, Dec., 1904, p. 618; Cassier's Mag., April, 1904, p. 506; Engng., Oct. 25, 1901, p. 595; Eng. News, May 22, 1902, p. 408; Eng. Rec., April 5, 1902, p. 322, June 10, 1905; Jour. Amer. Chem. Soc., Nov., 1893, p. 610; Jour. Soc. Chem. Ind., Aug., 1901, p. 828; Power, Nov., 1900, p. 7, Sept., 1902, p. 33, Nov., 1904, p. 693; R.R. Gaz., Aug. 24, 1900, p. 568; Elec. Rev., Nov. 12, 1904; Engr. U.S., Oct. 15, 1903, Jan. 1, 1906; Elec. World, Sept. 1, 1906.

Water-Softening Plants. —*Water-Softening Processes:* Prac. Engr. U.S., Mar., 1910. *Four Systems of Softening Water for Industrial Purposes:* Eng. News, July 2, 1903, p. 4. *An Inquiry into the Working of Various Water Softeners:* Inst. of Mech. Engrs., Dec., 1903. *Report on Soft Water for Locomotive Plants:* Eng. News, March 17, 1904. *The Development of Water Purification in the U.S.:* R.R. Gaz., 38-19. *General Information on Water Softening:* Eng. News, May 26, 1904, p. 500, 508, June 2, 1904, p. 530; Eng. Rec., Oct. 24, 1903, p. 483; Loco. Engng., Nov., 1903, p. 501; Elec. Engr., Lond., April 21, 1905; Ir. and Coal Tr. Rev., Sept. 1, 1905; Jour. W. Soc. Engrs., Dec., 1905; Eng. and Min. Jour., Dec. 2, 1905; Am. Engr. and R.R. Jour., Jan., 1905.

Harris-Anderson Water Softener: Engng., Aug. 15, 1902, July 10, 1903.

Kennicott Water Softeners: Am. Elecn., Nov., 1902, p. 545; Eng. News, May 15, 1902, p. 386; Eng. Rec., May 3, 1902, p. 419; St. Ry. Jour., April 2, 1904, p. 545.

Burt Continuous Water-Softening Process: Eng. News, Sept., 15, 1904, p. 238; Engr. U.S., July 15, 1905, p. 426.

Bachman Method of Water Purification: St. Ry. Review, May 15, 1900, p. 282.

We-Fu-Go and Scaife Systems: St. Ry. Rev., Oct., 1901, p. 771; Engr. U.S., Jan. 1, 1903, p. 90.

Holmes System of Water Purification: Power, April, 1905, p. 248.

The American Water Purifier and Softener: Eng. U.S., Aug. 1, 1904, p. 551.

248. Economy of Preheating Feed Water. — Although a feed water heater acts to some extent as a purifier its primary function is that of heating the feed water. Generally speaking, for every 10 degrees that the feed water is heated there is a gain in heat of 1 per cent and a corresponding saving of coal, if the heat which warms the feed water would otherwise be wasted. Again, the smaller the difference in temperature between the steam and the feed water the less will be the strain on the boiler shell due to unequal expansion and contraction, an item of no small consequence.

If λ represents the total heat of one pound of steam above 32 degrees F., t_0 the temperature of the cold water, and t the temperature of the water leaving the heater, then S, the per cent gain in heat due to heating the feed water, may be expressed

$$S = 100 \frac{(t - t_0)}{\lambda - (t_0 - 32)} \cdot \qquad (104)$$

The expression is not theoretically correct, since it assumes a constant value of unity for the specific heat, whereas the specific heat varies with the temperature. The variation is so slight, however, that it may be neglected for all practical purposes.

Example: Steam pressure 100 pounds gauge; temperature of water entering heater 80 degrees F.; temperature of water leaving heater 210 degrees F. Required, saving due to heating the feed water.

Here λ (from steam tables) is 1185, $t_0 = 80$, $t = 210$.

$$S = 100 \frac{(210 - 80)}{1185 - (80 - 32)}$$
$$= 11.42 \text{ per cent.}$$

This formula gives the *thermal* saving only, and the first cost of the heater, interest, depreciation, attendance, and repairs must be taken into consideration before the *net* saving measured in dollars and cents is ascertained. In the average installation the net saving is a substantial one.

Table 62 based upon formula (104) may be used in determining the percentages of saving due to the increase in feed-water temperature.

TABLE 62.

PERCENTAGE OF SAVING FOR EACH DEGREE OF INCREASE IN TEMPERATURE OF FEED WATER.

Initial Temp. of Feed.	Boiler Pressure above Atmosphere.										
	0	20	40	60	80	100	120	140	160	180	200
32	.0872	.0861	.0855	.0851	.0847	.0844	.0841	.0839	.0837	.0835	.0833
40	.0878	.0867	.0861	.0856	.0853	.0850	.0847	.0845	.0843	.0841	.0839
50	.0886	.0875	.0868	.0864	.0860	.0857	.0854	.0852	.0850	.0848	.0846
60	.0894	.0883	.0876	.0872	.0867	.0864	.0862	.0859	.0856	.0855	.0853
70	.0902	.0890	.0884	.0879	.0875	.0872	.0869	.0867	.0864	.0862	.0860
80	.0910	.0898	.0891	.0887	.0883	.0879	.0877	.0874	.0872	.0870	.0868
90	.0919	.0907	.0900	.0895	.0888	.0887	.0884	.0883	.0879	.0877	.0875
100	.0927	.0915	.0908	.0903	.0899	.0895	.0892	.0890	.0887	.0885	.0883
110	.0936	.0923	.0916	.0911	.0907	.0903	.0900	.0898	.0895	.0893	.0891
120	.0945	.0932	.0925	.0919	.0915	.0911	.0908	.0906	.0903	.0901	.0899
130	.0954	.0941	.0934	.0928	.0924	.0920	.0917	.0914	.0912	.0909	.0907
140	.0963	.0950	.0943	.0937	.0932	.0929	.0925	.0923	.0920	.0918	.0916
150	.0973	.0959	.0951	.0946	.0941	.0937	.0934	.0931	.0929	.0926	.0924
160	.0982	.0968	.0961	.0955	.0950	.0946	.0943	.0940	.0937	.0935	.0933
170	.0992	.0978	.0970	.0964	.0959	.0955	.0952	.0949	.0946	.0944	.0941
180	.1002	.0988	.0981	.0973	.0969	.0965	.0961	.0958	.0955	.0953	.0951
190	.1012	.0998	.0989	.0983	.0978	.0974	.0971	.0968	.0964	.0962	.0960
200	.1022	.1008	.0999	.0993	.0988	.0984	.0980	.0977	.0974	.0972	.0969
210	.1033	.1018	.1009	.1003	.0998	.0994	.0990	.0987	.0984	.0981	.0979
220	......	.1029	.1019	.1013	.1008	.1004	.1000	.0997	.0994	.0991	.0989
230	......	.1039	.1031	.1024	.1018	.1012	.1010	.1007	.1003	.1001	.0999
240	......	.1050	.1041	.1034	.1029	.1024	.1020	.1017	.1014	.1011	.1009
250	......	.1062	.1052	.1045	.1040	.1035	.1031	.1027	.1025	.1022	.1019

Multiply the factor in the table corresponding to any given initial temperature of feed water and boiler pressure by the total rise in feed-water temperature; the product will be the percentage of saving.

Feed Water Heating. — How Should Feed Water be Heated? — Power, July, 1907, p. 456; *Feed Water Heating:* Engr. U.S., Jan. 1, 1906, p. 8, Aug. 15, 1904, p. 15; St. Ry. Jour., July 22, 1905, p. 145; Am. Elecn., Dec., 1904, p. 570; Am. Elecn., Nov., 1904; Engr., Lond., July 28, 1905.

249. Classification of Feed-Water Heaters. — Feed-water heaters may be classified according to the *source* of heat, as

1. *Exhaust steam,* in which the heat is received from the exhaust of engines, pumps, etc.

2. *Flue gas,* in which the waste chimney gases are the source of the heat.

3. *Live steam purifiers,* or those using steam at boiler pressures; or according to the method of heat *transmission,* as

1. *Open heaters*, in which the steam and feed water mingle and the steam in condensing gives up its heat directly to the water.

2. *Closed heaters*, in which the steam and water are in separate chambers and the steam gives up its heat to the water by conduction.

Heaters may also be classified according to the pressure of the heating steam, as

1. *Vacuum* or *primary*, in which the pressure is less than atmospheric and applies particularly to heaters utilizing the exhaust of condensing engines. These are always of the closed type. Open heaters in which the pressure is less than atmospheric are not usually classed as vacuum heaters.

2. *Atmospheric* or *secondary*, in which the pressure is atmospheric or, literally, that corresponding to the back pressure on the engines and pumps.

3. *Pressure*, in which the pressure corresponds to that in the boiler and in which the heat is used primarily for purifying purposes.

Heaters may be still further classified as

1. *Induced*, in which only such steam is admitted as is induced by its condensation. That is, the feed water condenses the steam. This creates a partial vacuum which draws in more steam.

2. *Through*, in which all the steam is forced through the heater irrespective of condensation.

CLASSIFICATION OF A FEW TYPICAL HEATERS.

Exhaust steam	Open	Atmospheric	Cochrane Hoppes Stillwell Webster	
	Closed	Atmospheric Vacuum or pressure	Wainwright Wheeler	Water Tube
			Otis Berryman	Steam Tube
Flue Gas			Green American Sturtevant	
Live Steam	Open	Pressure	Hoppes Baragwanath	

250. Open Heaters. — Fig. 239 gives a sectional view of a Cochrane special feed heater and receiver and is a typical example of an open heater. Exhaust steam enters the heater through a fluted oil separator as indicated, and passes out at the top, while the oily drips are automatically drained to waste by a suitable ventilated float. The feed water enters through an automatic valve and is distributed over

a series of copper trays so arranged and constructed that the water is forced to fall in a finely divided stream before reaching the reservoir in the bottom. The steam coming in contact with the water particles gives up latent heat and condenses. Much of the scale-forming element is deposited on the surface of the trays, from which it is readily removed. The suspended matter is eliminated by a coke filter in the

Fig. 239. Cochrane Special Heater and Receiver.

bottom of the chamber, and the floating impurities are decanted by a skimmer or overflow weir. The particular heater shown in the illustration is especially designed for use in a steam-heating plant; i.e., besides performing all the functions of an open heater, it provides for the reception and heating of the condensation returned to it from the heating system.

Fig. 240 gives a sectional view of a Webster "star vacuum" heater. Water enters the heater through balanced valve F, which is controlled by float E, and is deflected over a series of perforated copper trays T, T. Exhaust steam enters at A, passes through oil filter S, and, mingling with the finely divided streams of water, gives up its latent heat and is condensed. Only so much steam enters the heater as is condensed by the feed water. The condensed steam and feed water fall to the bottom

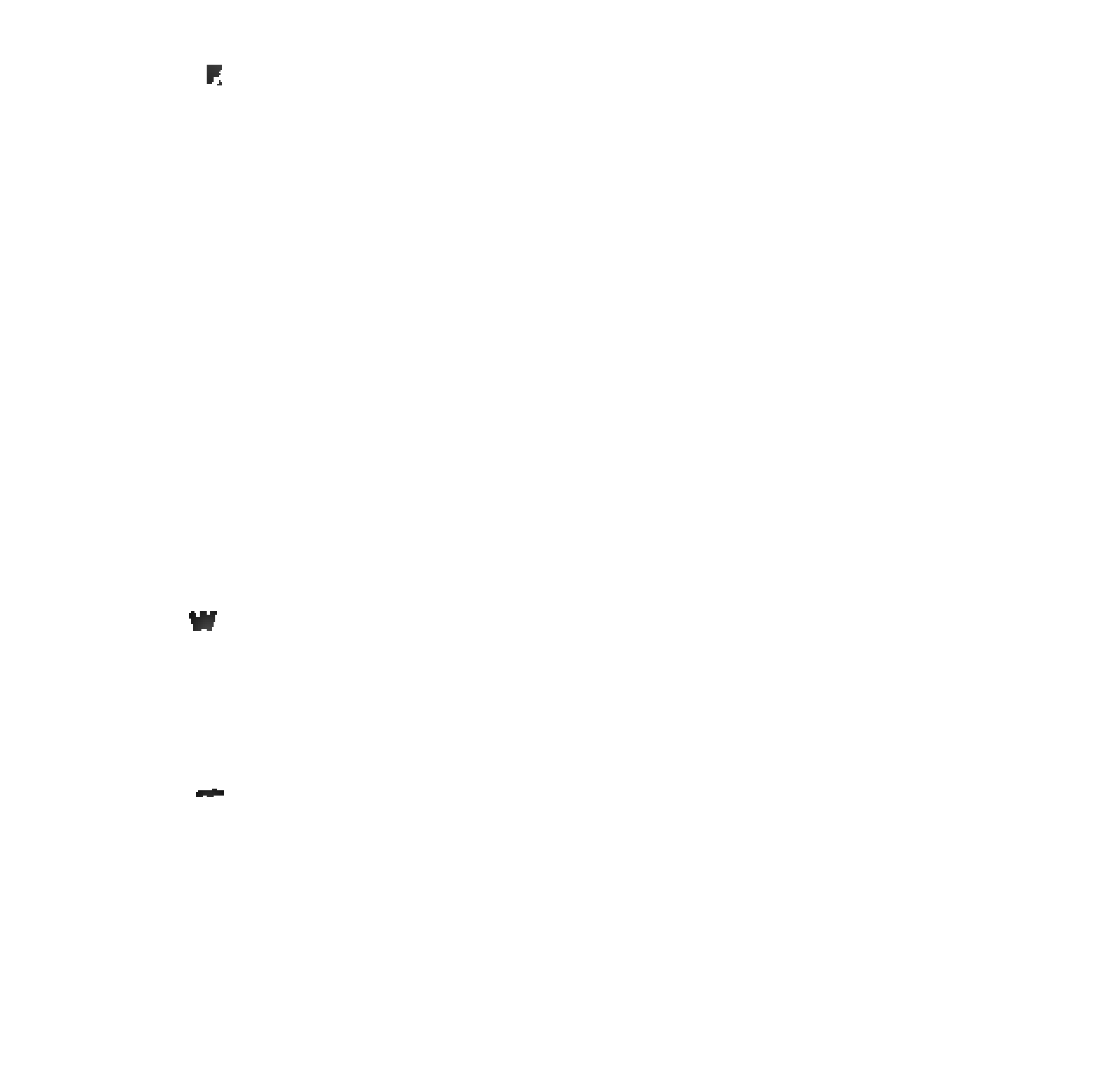


FIG. 240. Section Through Webster Heater.

of the upper chamber, maintaining a practically constant level WW. From this upper or heater chamber the water gravitates to the settling chamber at the bottom, through down-cast pipe CB. From the settling chamber the water rises through perforated screen M and filtering material P to the outlet O. A large portion of the scale-forming element is precipitated on the trays or collects in the settling chamber at the bottom.

Fig. 241 shows a section through a Hoppes open heater, illustrating the "pan" type. Exhaust steam enters at H, passes through oil filter O, and completely surrounds pans T, T. The feed water enters at B, and the rate of flow is regulated by valve F, which is controlled by a

C

FIG. 241. Hoppes Horizontal Feed-Water Heater.

suitable float in the lower part of the chamber. The water in flowing over the sides and bottoms of the pans comes in direct contact with the steam.

251. Combined Open Heater and Chemical Purifier. — Combined feed-water heaters and chemical purifiers are finding increased favor with engineers in many districts where the feed water is particularly bad. A description of the Webster combination will be found in Part II of the general catalogue issued by the Warren Webster Company, Camden, N.J. A description of the Cochrane-Sorge combined heater and chemical purifier will be found in the heater catalogue issued by the Harrison Safety Boiler Works, Philadelphia, Pa.

252. Temperatures in Open Heaters. — The temperature to which feed water is raised in an open heater may be determined as follows:

Let λ represent the total heat of steam corresponding to the pressure in the heater,

t_0 the temperature of the water entering heater,

t the temperature of the water leaving heater, and

S the ratio of exhaust steam to the feed water, by weight.

from 0.25 for good quality of water to 0.4 of the volume of the shell for muddy water, 0.33 being about the average. In the vertical type the settling chamber represents respectively 0.4 and 0.6 the volume of the shell with clear and muddy water. Filters occupy from 10 to 15 per cent of the volume of the shell in the horizontal type and from 15 to 20 per cent in the vertical type, the smaller percentage corresponding to clear water and the larger to muddy water or water containing a considerable quantity of impurities.

Open Heaters: Cassier's Mag., Aug., 1903, p. 33; Engr. U.S., Jan. 1, 1906, pp. 17, 78; St. Ry. Jour., Feb. 4, 1905, p. 227; Am. Elecn., Sept., 1905, p. 481.

255. Classification of Closed Heaters. — Closed heaters may be grouped into two classes:

1. Water tube, Fig. 242, and
2. Steam tube, Fig. 246.

Closed heaters, both water tube and steam tube may operate with:

1. *Parallel currents,* where the water and steam flow in the same direction, Fig. 242, or with
2. *Counter currents,* where the water and steam flow in opposite directions, Fig. 244.

FIG. 242. Goubert Single-Flow Closed Heater.

FIG. 243. Details of Expansion Joint, Goubert Heater.

Water-tube heaters may be still further classified as

1. *Single-flow,* in which the water flows through the heaters in one direction only, Fig. 242.

2. *Multi-flow*, in which the water flows back and forth a number of times, as in Fig. 244.

3. *Coil heater*, in which the water flows through one or more coils, as in Fig. 245.

256. Water-Tube, Closed Heaters. — Fig. 242 shows a section through a feed-water heater of the single-flow straight-tube type. The

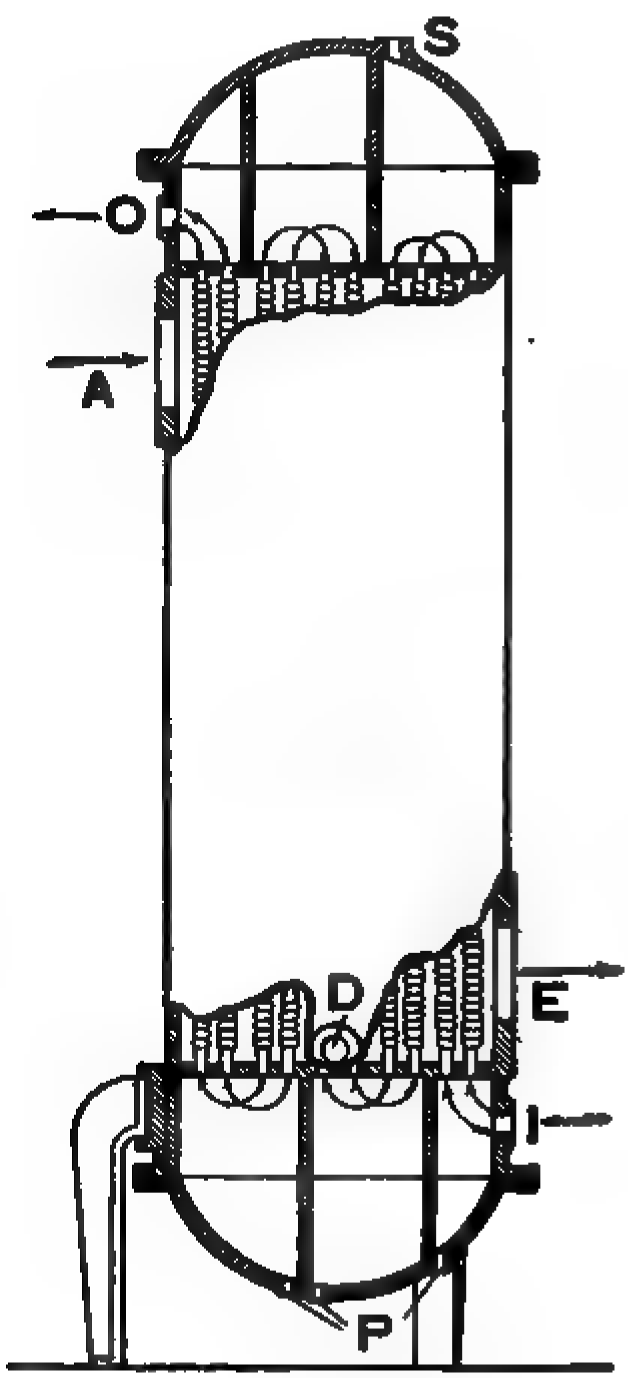


FIG. 244. Wainwright Multi-Flow Closed Heater.

FIG. 245. Typical Coil Heater.

tubes are of plain brass and the shell of cast iron. The tubes are expanded into the tube sheets by a roller expander. To provide for expansion the upper tube sheet and water chamber are secured to the main shell by means of a special expansion joint the details of which are shown in Fig. 243. *R* is a ring or gasket of soft annealed copper

and G, G two gaskets of special packing with brass wire cloth insertion. These gaskets form a flexible expansion joint between C and tube sheet D, so that the whole upper chamber, which is carried solely by the tubes, is free to move up and down as the tubes expand or contract under varying temperatures.

Fig. 244 shows a section through a Wainwright heater, illustrating the multi-flow water-tube type. The body of the heater is of cast iron, the tubes of corrugated copper. The water passes through the tubes and the steam surrounds them. The feed water and exhaust steam do not mingle, and hence the oil in the exhaust does not contaminate the water. The water chambers are divided into several compartments, as shown in the illustration, and the partitions are so arranged that the flow of feed water is directed back and forth through the various groups of tubes in succession. This arrangement gives a higher velocity of flow than the non-return type of heater, and therefore increases the rate of heat absorption. The mud and impurities settle at the bottom and are discharged through the mud blow-off. Such impurities as rise to the surface are removed by the surface blow-off. The tubes are corrugated to allow for expansion and at the same time to increase the transmission of heat. Referring to Fig. 244: Exhaust steam enters at A and leaves at E, and the portion which is condensed is drawn off at D. Feed water enters at I and is discharged at O. P, P are mud blow-offs and S is an opening for a safety valve. Table 66 gives results of tests showing the relative efficiencies of plain and corrugated tubes for various velocities.

Fig. 245 shows a partial section through a Harrisburg feed-water heater. This apparatus is a typical example of the coiled-tube heater. Three sets of concentric copper coils are brazed to gun-metal manifolds and supported by clamp stays as indicated in the illustration. Feed water enters the heater at the bottom manifold and passes through the coils to the feed outlet. The exhaust steam enters the heater at the bottom and surrounds the coils in its passage to the outlet at the top. The coils are designed to withstand a pressure of 600 pounds per square inch.

257. Steam-Tube, Closed Heaters. — Fig. 246 shows a section through an Otis heater, illustrating the steam-tube type. Here the exhaust steam passes *through* the tubes which are surrounded by the feed water. The exhaust steam enters at A, and passes down one section of tubes into the enlarged space of the water and oil separator O, in which the condensation and oil are deposited. From this chamber the steam passes up through the other section of tubes to outlet C, thus

passing twice through the entire length of the heater. The water enters at *E* and is discharged at *G*. *R* is the blow-off opening. The tubes are of seamless brass and are curved to allow for expansion. Condensed steam is withdrawn at *P*.

Fig. 247 shows a partial section through a Baragwanath steam jacketed steam-tube heater. Exhaust steam enters at *A*, passes up

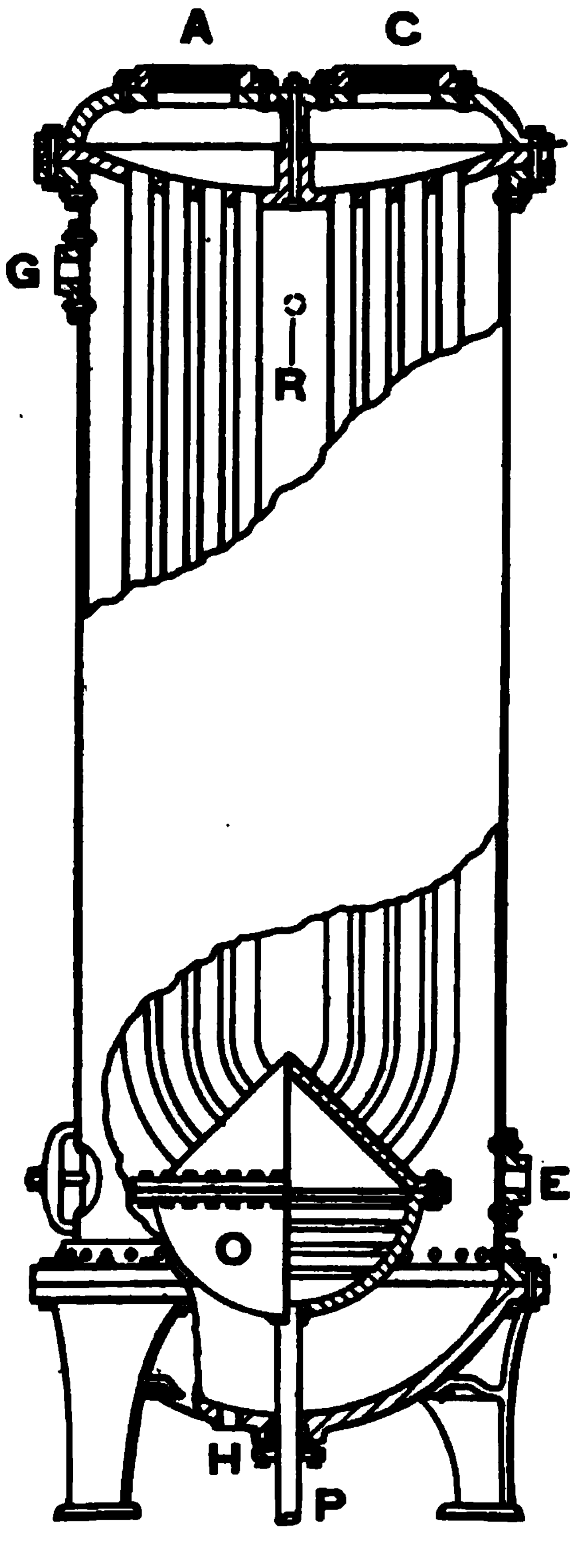


FIG. 246. Otis Steam-Tube Feed-Water Heater.

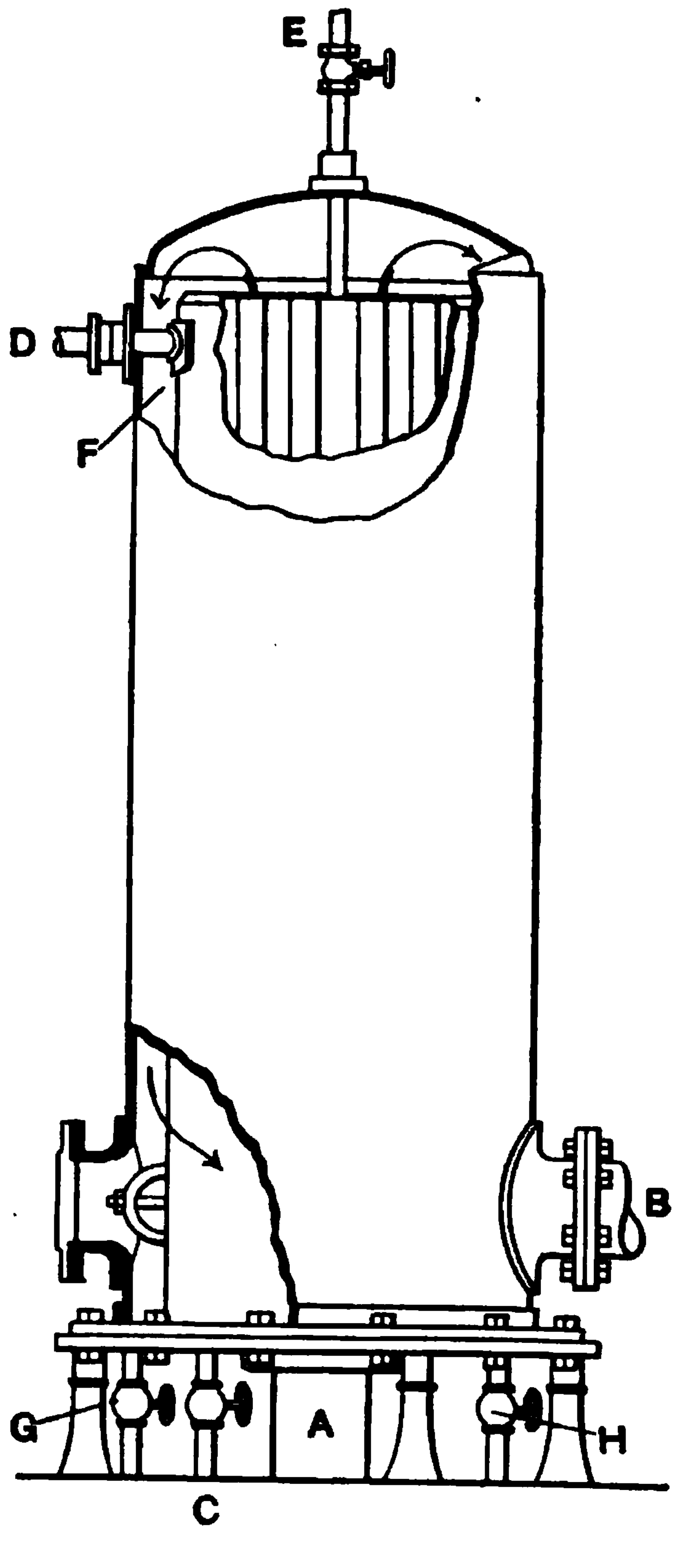


FIG. 247. Baragwanath Steam-Jacketed Feed-Water Heater.

through the tubes, returns down annular space *E* between the inner shell and jacket, and passes out at *B*. Feed water enters at *C* and leaves at *D*. *E* is the scum blow-off, *G* the heater drain, and *H* the jacket drain.

258. Heating Surface, Closed Heaters. — It is generally assumed that the transfer of heat between two bodies is directly proportional to the difference in temperature between them.

Let T_0 = temperature of the water entering the heater.
T_2 = temperature of the water leaving the heater.
T_s = temperature of the exhaust steam.
A = square feet of transmitting surface.
T = temperature of a unit of water t seconds after entering the heater.
h = B.T.U. absorbed per square foot per second per degree difference in temperature between the steam temperature T_s and the water temperature T.
t = time in seconds.
w = number of pounds of feed water per second.

Then $\frac{A}{w}$ = square feet of surface brought in contact with one pound of water per second,

and dT, the rate at which the temperature of the water is increasing at this instant, will be

$$dT = \frac{hA}{w}(T_s - T)\,dt. \tag{108}$$

$$\frac{dT}{T_s - T} = \frac{hA}{w}\,dt. \tag{109}$$

Integrating,

$$\int \frac{dT}{T_s - T} = \frac{hA}{w}\int dt. \tag{110}$$

$$\int_{T_0}^{T_2} \frac{dT}{T_s - T} = \frac{hA}{w}\int_0^t dt. \tag{111}$$

$$\log_e \frac{T_s - T_0}{T^s - T_2} = \frac{hAt}{w}. \tag{112}$$

Let W = number of pounds of feed water heated per hour.
U = B.T.U. transmitted to the feed water per square foot of surface per hour per degree difference in temperature.

Then (112) may be written

$$\log_e \frac{T_s - T_0}{T_s - T_2} = \frac{AU}{W}, \tag{113}$$

from which

$$A = \frac{W}{U}\log_e \frac{T_s - T_0}{T_s - T_2}. \tag{114}$$

Knowing the weight of water to be heated, the temperature of the steam, the desired temperature of the feed water, and the coefficient of heat transmission, U, this equation enables one to determine the area of heating surface required for the given conditions. Since the extent of heating surface increases rapidly as T_2 approaches T_s, and becomes infinity for $T_2 = T_s$, it is desirable to limit T_2 to some practical figure. An average maximum for $T_2 = T_s - 4$.

Table 64 has been calculated from this formula and gives the square feet of heating surface necessary to heat 1000 pounds of water per hour for different ranges in temperature.

Mean Temperature Difference.

If we let d = average temperature difference between the steam and feed water, then

$$AUd = \text{heat given out by the steam per hour.}$$

$$W(T_2 - T_0) = \text{heat absorbed by the feed water per hour.}$$

$$AUd = W(T_2 - T_0). \tag{115}$$

$$d = \frac{W(T_2 - T_0)}{AU}. \tag{116}$$

From (113),

$$\frac{AU}{W} = \log_e \frac{T_s - T_0}{T_s - T_2}. \tag{117}$$

Therefore

$$d = \frac{T_2 - T_0}{\log_e \dfrac{T_s - T_0}{T_s - T_2}}. \tag{118}$$

Table 65 has been calculated from formula (118) and gives the mean temperature difference for various conditions of operation.

The arithmetic mean temperature difference d_1 may be taken with safety for the average heater problem and has the advantage of simplicity.

$$d_1 = T_s - \frac{T_0 + T_2}{2}. \tag{119}$$

Closed heaters are sometimes rated on the basis of $\frac{1}{3}$ square foot of heating surface per horse power, i.e., a heater with 500 square feet of heating surface would be rated at 1500 horse power.

259. Heat Transmission in Closed Heaters. — Table 66 gives the results of a series of tests on the absorption of heat by water passing through brass and copper tubes surrounded by steam. The curves in Fig. 248 were plotted from the data given in this table. An inspection of the table and the curves will show that the absorption of heat per square foot of surface per degree difference in temperature varies with

TABLE 64.

SQUARE FEET OF HEATING SURFACE REQUIRED TO HEAT 1000 POUNDS OF WATER PER HOUR.
$U = 350$.

Vacuum Heaters between Engine and Condenser. Final Temperature of Feed Water.

Initial Temperature of Feed Water, T_0.	24″ Vacuum. Temperature 141° F.						25″ Vacuum. $T_s = 134°$ F.			
	105	110	115	120	125	130	110	115	120	125
40	2.93	3.36	3.86	4.50	5.22	6.29	3.93	4.65	5.58	7.01
50	2.64	3.29	3.57	4.15	4.93	6.01	3.65	4.36	5.28	6.65
60	2.29	2.93	3.22	3.86	4.58	5.65	3.29	4.01	4.93	6.29
70	1.93	2.50	2.86	3.43	4.22	5.29	3.07	3.58	4.57	5.86
80	1.50	2.07	2.43	3.01	3.72	4.86	2.36	3.07	4.01	5.36
T_0.	26″ Vac. $T_s = 125°$.				27″ Vac. $T_s = 114°$.			28″ Vac. $T_s = 100°$.		
	Final Temperature of the Feed Water.									
	105	110	115	120	90	100	105	70	80	90
40	4.43	4.93	6.07	8.18	3.22	4.72	6.01	1.93	3.14	5.08
50	3.92	4.57	5.72	7.73	2.79	4.36	5.58	1.43	2.57	4.57
60	3.36	4.15	5.36	7.51	2.29	3.86	5.08	.78	1.93	3.93
70	2.86	3.65	4.79	6.86	1.71	3.22	4.50	......	1.14	3.14
80	2.29	3.07	4.28	6.28	.86	2.21	3.79	......	......	1.93

Atmospheric Heaters. Atmospheric Pressure. Temp. 212° F. Final Temperature of Feed Water.

192	196	200	204	208	210	Initial Temperature of Feed Water, T_0.
6.01	6.65	7.58	8.73	10.72	12.74	40
5.94	6.58	7.44	8.58	10.51	12.51	50
5.79	6.44	7.15	8.36	10.38	12.30	60
5.58	6.22	7.01	8.23	10.15	12.15	70
5.37	6.01	6.87	8.00	9.94	11.85	80
5.15	5.79	6.57	7.80	9.80	11.79	90
4.93	5.51	6.36	7.51	9.52	11.50	100
4.79	5.46	6.22	7.37	9.37	11.37	105
4.65	5.28	6.15	7.22	9.23	11.15	110
4.50	5.15	5.93	7.08	9.08	11.07	115
4.36	4.93	5.79	6.93	8.95	10.94	120
4.15	4.79	5.64	6.79	8.80	10.80	125
3.86	4.65	5.44	6.57	8.65	10.58	130

TABLE 65.

DEGREES OF DIFFERENCE BETWEEN STEAM TEMPERATURE AND ACTUAL AVERAGE TEMPERATURE OF FEED WATER.

Vacuum Heaters between Engine and Condenser.

Initial Temperature of Water, T_0.	24″ Vacuum. Temperature 141° F. Final Temperature of Water.						25″ Vacuum. Temp. 134° F. Final Temperature of Water.			
	105	110	115	120	125	130	110	115	120	125
40	62.9	59.3	55.3	50.9	46.2	40.6	50.1	45.7	40.6	34.6
50	59.2	55.6	51.9	47.7	43.1	37.8	46.7	42.4	37.7	32.1
60	55.5	52.1	48.4	44.4	40.1	35.1	43.2	39.3	34.7	29.4
70	51.6	48.2	44.8	41.1	36.9	32.1	39.7	35.9	31.2	26.6
80	47.6	44.2	41.0	37.5	33.5	29.2	35.9	32.4	28.4	23.8

Initial Temperature of Water, T_0.	26″ Vac. Temp. 125° F. Final Temp. of Water.				27″ Vac. Temp. 114°. Final Temp. of Water.			28″ Vac. Temp. 100°. Final Temp. of Water.		
	105	110	115	120	90	100	105	70	80	90
40	42.0	40.4	35.0	28.3	44.4	36.0	30.8	43.2	36.3	27.9
50	39.9	37.3	32.3	25.9	40.8	32.8	28.0	39.1	32.7	24.8
60	38.2	34.1	29.4	23.4	36.9	29.6	25.1	34.7	28.8	21.6
70	34.3	30.8	26.9	20.7	33.0	26.2	22.1	......	24.7	18.1
80	30.8	27.3	23.3	18.1	28.7	22.5	18.7	......	......	14.4

Atmospheric Heaters.

Atmospheric Pressure. Temp. 212° F. Final Temperature of Water.						Initial Temperature of Water T_0.
192	196	200	204	208	210	
70.6	65.7	60.1	53.5	44.8	38.2	40
67.9	63.1	57.6	51.2	42.8	36.4	50
65.1	60.4	55.2	48.9	40.7	34.7	60
62.2	57.7	52.6	46.6	38.7	32.9	70
59.4	54.9	50.0	44.2	36.6	31.0	80
56.4	52.3	47.4	41.8	34.4	29.1	90
53.1	49.3	44.7	39.3	32.4	27.3	100
51.9	47.9	43.4	38.2	31.4	26.4	105
50.3	46.4	42.1	36.9	30.2	25.5	110
48.8	45.0	40.6	35.7	29.2	24.5	115
47.2	43.5	39.2	34.4	28.0	23.5	120
45.6	41.9	37.8	33.1	26.9	22.5	125
43.9	40.3	36.4	31.7	25.8	21.5	130

the velocity of the water and the material and character of the tubes. Increasing the velocity of the water passing through the heater increases the rate of heat transmission and thereby renders the heating surface more effective. In order to employ moderately high velocities and at the same time allow sufficient time in which to raise the temperature to a maximum, the tubes should be as long as practicable and of small diameters. Other things being equal, a heater containing a large number of tubes of small diameter is more economical than one containing a small number of large tubes. It is important to proportion the heater according to the amount of water to be heated and the maximum temperature to which the water must be raised. In designing a heater, then, the maximum amount of heat to be transmitted per degree difference in temperature per hour per square foot should be assumed, and the velocity of the water made such that it is capable of absorbing this amount. A good average figure for multi-flow heaters is $U = 250$ B.T.U. for plain brass or copper tubes and $U = 300$ B.T.U. for corrugated tubes with a water velocity of 50 feet per minute; for single-flow heaters, $U = 175$ (for plain brass) with a water velocity of 12.5 feet per minute and for coil heaters $U = 300$ (copper) with a water velocity of 150 feet per minute. These figures are for water-tube heaters only. For steam-tube heaters (iron tubes) a good average figure is $U = 120$.

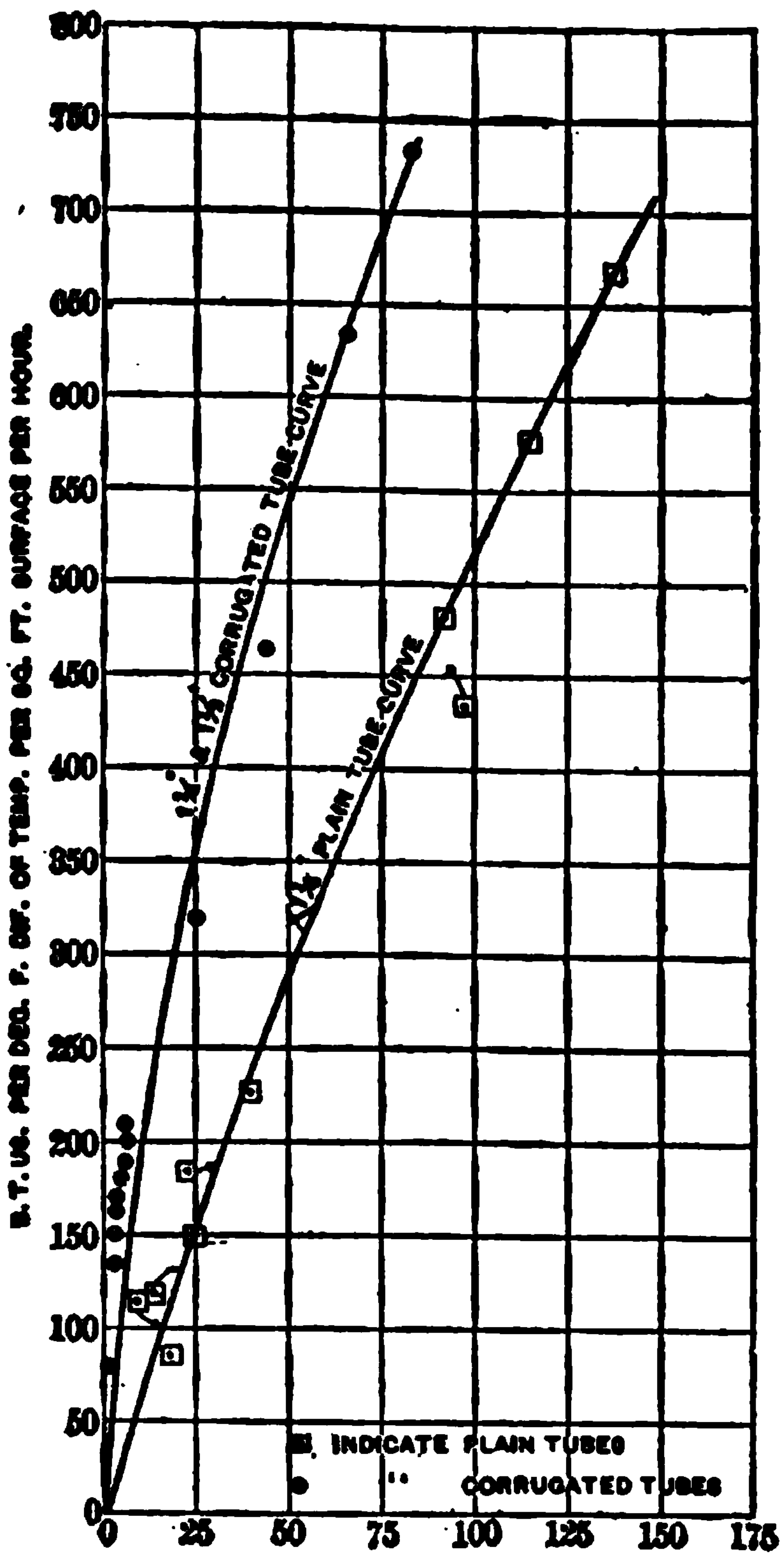


Fig. 248.

Experiments show that heaters and condensers operating with counter-currents are more efficient and are capable of obtaining a higher final temperature than those operating with parallel currents.

Example: Determine the size of vacuum and atmospheric heaters for a condensing plant of 1200 I.H.P. Engines use 20 pounds of steam per I.H.P. hour; auxiliaries use the equivalent of 10 per cent of the main engine steam; vacuum 25 inches referred to 30-inch

barometer; feed water, $T_0 = 50$ degrees; temperature of hot well, $T_2 = 110$ degrees; coefficient of heat transmission, $U = 300$ B.T.U.

Vacuum or Primary Heater.

Feed water for main engines,

$$20 \times 1200 = 24{,}000 \text{ pounds per hour.}$$

Feed water used by auxiliaries,

$$10 \text{ per cent of } 24{,}000 = 2400 \text{ pounds per hour.}$$

Total feed,

$$W = 24{,}000 + 2400 = 26{,}400 \text{ pounds per hour.}$$

From formula (114),

$$A = \frac{W}{U} \log_e \frac{T^s - T_0}{T_s - T_2}$$
$$= \frac{26{,}400}{300} \log_e \frac{134 - 50}{134 - 110}$$
$$= 110 \text{ square feet.}$$

On the basis of $\frac{1}{3}$ square foot of surface per horse power the rating of this heater will be

$$110 \times 3 = 330 \text{ horse power.}$$

Atmospheric or Secondary Heater.

The temperature of the feed water leaving the atmospheric heater, formula (105), will be

$$t = \frac{t_0 + 0.9\, S\, (\lambda + 32)}{1 + 0.9\, S},$$

where $S = .10,\ t_0 = 110 \text{ degrees},\ \lambda = 1146 \text{ B.T.U.}$

whence $$t = \frac{110 + 0.9 \times 0.10\,(1146 + 32)}{1 + 0.9 \times 0.10}$$
$$= 198 \text{ degrees.}$$

The required surface is

$$A = \frac{W}{U} \log_e \frac{T_s - T_0}{T_s - T_2},$$

where $T_s = 212,\ T_0 = 110,\ T_2 = 198,$

whence $$A = \frac{26{,}400}{300} \log_e \frac{212 - 110}{212 - 198}$$
$$= 175 \text{ square feet.}$$

The horse-power rating will be

$$175 \times 3 = 525 \text{ horse power.}$$

TABLE 66.

ABSORPTION OF HEAT BY WATER PASSING THROUGH BRASS AND COPPER TUBES SURROUNDED BY STEAM.

(*Power*, December, 1901.)

No. of Experiment.	Kind of Tubes.	Material of Tubes.	Number of Tubes.	Length of Tubes.	Outside Diameter of Tubes.	Gauge of Tubes.	Position of Tubes.	Direction of Flow of Water.	Square Feet of Heating Surface.	Temp. of Steam, Degrees F.	Degrees F. of Inlet Water.	Degrees F. of Outlet Water.	Pounds of Water per Hour.	Pounds of Water per Hour per Single Tube.	Velocity of Water in Feet per Minute.	Total Number of B.T.U.'s Absorbed per Hour.	B.T.U.'s Absorbed per Sq. Ft. of Surface per Degree Dif. of Temp. per Hour.	Experimenter.
1	P	Cu	1	18"	1/2"	19	Vert.	Up	.1964	212	47.8	96.7	175.875	175.875	48.24	8600.287	314	Stanton
2	P	Cu	1	18	1/2	19	Vert.	Up	.1964	212	47.8	78.0	356.56	356.56	97.8	10768.112	376	Stanton
3	P	Cu	1	18	1/2	19	Vert.	Up	.1964	212	46.7	73.2	463.75	463.75	127.2	12289.375	411	Stanton
4	P	Cu	1	18	1/2	19	Vert.	Up	.1964	212	54.3	74.1	805.625	805.625	248.4	15951.375	537	Stanton
5	P	Cu	1	18	1/2	19	Vert.	Down	.1964	212	47.8	110.3	175.875	175.875	48.24	10992.187	426	Stanton
6	P	Cu	1	18	1/2	19	Vert.	Down	.1964	212	47.8	83	356.56	356.56	97.8	12550.912	434	Stanton
7	P	Cu	1	18	1/2	19	Vert.	Down	.1964	212	46.7	77.5	463.75	463.75	127.2	14283.500	489	Stanton
8	P	Cu	1	18	1/2	19	Vert.	Down	.1964	212	54.3	74.9	805.625	805.625	248.4	16595.875	573	Stanton
9	P	B	1	65	3/4	18	Vert.	Down	1.065	255	58	140	702	702	81	57564	353.6	Nichols
10	P	B	1	65	3/4	18	Vert.	Down	1.065	255	58	93.47	2421	2421	278	85884.975	452	Nichols
11	P	B	1	65	3/4	18	Vert.	Down	1.065	256	58	84.95	3393	3393	390	91441.35	473	Nichols
12	P	B	1	65	3/4	18	Horiz.	Horiz.	1.065	253	57.7	165	675.14	675	78	72393.75	505	Nichols
13	P	B	1	65	3/4	18	Horiz.	Horiz.	1.065	253	58	101	2670	2670	307	114810	619	Nichols
14	P	B	1	65	3/4	18	Horiz.	Horiz.	1.065	254	58	94.5	3612	3612	415	131838	701	Nichols
15	P	Cu	2	240	1 1/4	18	Horiz.	Horiz.	13.09	212	74	193	1216	608	22.48	144704	184	Hepburn
16	P	Cu	2	240	1 1/4	18	Horiz.	Horiz.	13.09	212	74	180	1342	671	24.81	142252	149	Hepburn
17	P	Cu	6	240	1 1/4	18	Horiz.	Horiz.	39.27	212	74	170	1870	478.33	17.69	275520	86.9	Hepburn
18	P	Cu	6	240	1 1/4	18	Horiz.	Horiz.	39.27	212	72	195	2292	362	13.38	281916	123	Hepburn
19	P	Cu	6	240	1 1/4	18	Horiz.	Horiz.	39.27	211	74	204	1548	258	9.54	201240	117.1	Hepburn
20	P	Cu	1	240	1 1/4	18	Horiz.	Horiz.	6.045	217	71	170	3082	3082	113.95	305118	578.4	Hepburn
21	P	Cu	1	240	1 1/4	18	Horiz.	Horiz.	6.045	216	72	172	2464	2462	91.10	246400	482.9	Hepburn
22	P	Cu	1	240	1 1/4	18	Horiz.	Horiz.	6.045	216	71	162	3708	3708	137.1	337428	669.6	Hepburn
23	P	Cu	1	240	1 1/4	18	Horiz.	Horiz.	6.045	214	72	174	1084	1084	40.08	110568	227.2	Hepburn

24	Cor.	Cu	6	240	1¼	18	Horiz.	Horiz.	39.27	211	41	204	3930	655	24.2	640590	318.1	Hepburn
25	Cor.	Cu	6	240	1¼	18	Horiz.	Horiz.	39.27	211.5	40	204	4006	667.67	24.69	656984	319.7	Hepburn
26	Cor.	Cu	2	240	1¼	18	Horiz.	Horiz.	13.09	215	67	195	4440	2220	82.08	568320	735	Hepburn
27	Cor.	Cu	2	240	1¼	18	Horiz.	Horiz.	13.09	211	66	200	3530	1765	65.26	473020	634	Hepburn
28	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	207	2798.7	116.61	2.91	405811.5	175.2	Hildreth
29	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	201¼	3965	165.21	4.13	552126.25	202.8	Hildreth
30	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	193	5222.9	217.62	5.43	684199.9	215.2	Hildreth
31	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	194¼	5161.7	215.07	5.36	682634.825	218.7	Hildreth
32	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	208½	2558.4	106.6	2.76	374805.6	170.1	Hildreth
33	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	206½	3068.5	127.86	3.19	443398.25	188.2	Hildreth
34	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	201¼	3992.6	166.36	4.16	555969.55	204	Hildreth
35	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	195¼	4950.2	206.26	5.14	659614.15	215.1	Hildreth
36	Cor.	Cu	24	60	1½	18	Vert.	Up.	47.12	215	62	197	4805.2	200.22	4.99	648702	218.2	Hildreth
37	Cor.	Cu	36	74	1½	18	Vert.	Up.	87.18	215	62	211¼	1214.9	33.75	.84	181323.825	51.6	Hildreth
38	Cor.	Cu	36	74	1½	18	Vert.	Up.	87.18	215	62	211	3295.6	91.54	2.28	491044.4	137.6	Hildreth
39	Cor.	Cu	36	74	1½	18	Vert.	Up.	87.18	215	62	210	3807.1	105.75	2.64	563450.8	149.3	Hildreth
40	Cor.	Cu	36	74	1½	18	Vert.	Up.	87.18	215	62	208	4467.5	124.1	3.09	652255	164	Hildreth
41	Cor.	Cu	36	74	1½	18	Vert.	Up.	87.18	215	62	205	5878.6	163.3	4.08	840693.8	184.1	Hildreth
42	Cor.	Cu	36	74	1½	18	Vert.	Up.	87.18	215	62	203	6387.4	177.43	4.43	900623.4	184.2	Hildreth
43	Cor.	Cu	36	74	1½	18	Vert.	Up.	87.18	215	62	200¾	7128	198	4.96	989010	190.5	Hildreth
44	Cor.	Cu	36	74	1½	18	Vert.	Up.	87.18	215	62	198	7869.2	218.6	5.46	1070211.2	197.2	Hildreth
45	Cor.	Cu	3	296	1½	18	Horiz.	Horiz.	29	219.45	45	211.8	4111	1370.3	34.18	685714.8	444	Richter
46	Cor.	Cu	3	296	1½	18	Horiz.	Horiz.	29	229.3	43	215.2	5231	1743.7	43.5	900778.2	465	Richter
47	Cor.	Cu	3	296	1½	18	Horiz.	Horiz.	29	233	45	227	3936	1312	32.73	716352	466	Richter

P = Plain Tube. Cor. = Corrugated Tube. Cu = Copper. B = Brass.

260. Open *vs.* Closed Heaters. — Open and closed heaters have their respective advantages and a careful study of the various influencing conditions is necessary for an intelligent choice. The following parallel comparison brings out a few of the distinguishing features:

Open Heater.	Closed Heater.
Efficiency.	
With sufficient exhaust steam for heating, the feed water may reach the same temperature as the steam.	The maximum temperature of the feed water will always be 2 degrees or more lower than the temperature of the steam.
Scale and oil do not affect the heat transmission.	Scale and oil deposit on the tubes and the heat transmission is lowered.
Pressures.	
It is not ordinarily subjected to much more than atmospheric pressure.	The water pressure is slightly greater than that in the boiler when placed on the pressure side of the pump as is customary.
Safety.	
Sticking of the back pressure valve may cause it to " blow up " if provision is not made for such an emergency.	It will safely withstand any pressure likely to occur.
Purification.	
Since the exhaust steam and feed water mingle, provision must be made for removing the oil from the steam.	Oil does not come in contact with the feed water.
Scale and other impurities precipitated in the heater are readily removed.	Scale is removed with difficulty.
Location.	
Must always be placed above the pump suction and on the suction side.	May be placed anywhere on the pressure side of the pump.
Pumps.	
With supply under suction two pumps are necessary and one must handle hot water.	One cold-water pump is necessary.
Adaptability.	
Particularly adaptable for heating systems where it is desired to pipe the " returns " direct to heater.	All vacuum or primary heaters are necessarily of this type.

261. "Through" Heaters. — Fig. 249 shows a typical installation of a through heater in a non-condensing plant.

It is evident that *all* the steam must pass *through* the heater. Now, one pound of exhaust steam in condensing gives up approximately

Fig. 249. Open Heater Connected as a "Through" Heater. Non-Condensing Plant.

1000 B.T.U. Hence, if the initial temperature of the feed water is 50 degrees and the final temperature 210, the engine furnishes $\frac{1000}{210 - 50} = 6.26$, say six times the quantity necessary for heating the feed water to a maximum. Therefore the area of the pipe supplying the heater with steam need be but one-sixth that of the main exhaust. With the heater connected as in Fig. 249 the connections must necessarily be the same size as the exhaust pipe.

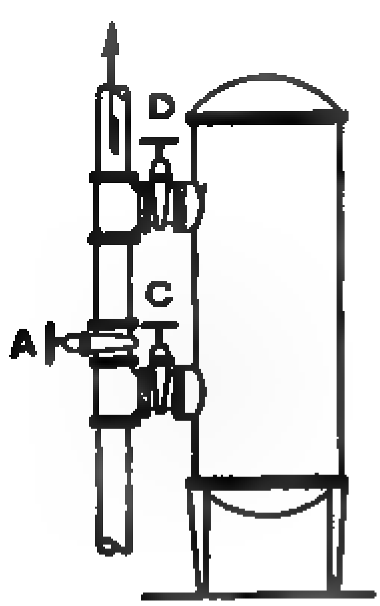


Fig. 250.

With this arrangement the heater cannot be "cut out" while the engine is in operation and hence it is not adapted for plants working continuously. For the purpose of cutting out a heater while the plant is in operation a through heater may be by-passed as in Fig. 250. Advantage may be taken here of the permissible reduction in the size of pipes and fittings, i.e., valves, etc., at *C* and *D* need be but one-half the size of those at *A*. This reduction in size may prove to be a considerable item in large installations.

262. Induced Heaters. — Fig. 251 shows a typical installation of an induced heater in a non-condensing plant and Fig. 252 an induced primary heater in a condensing plant.

FIG. 251. Open Heater Connected as an "Induced" Heater. Non-Condensing Plant.

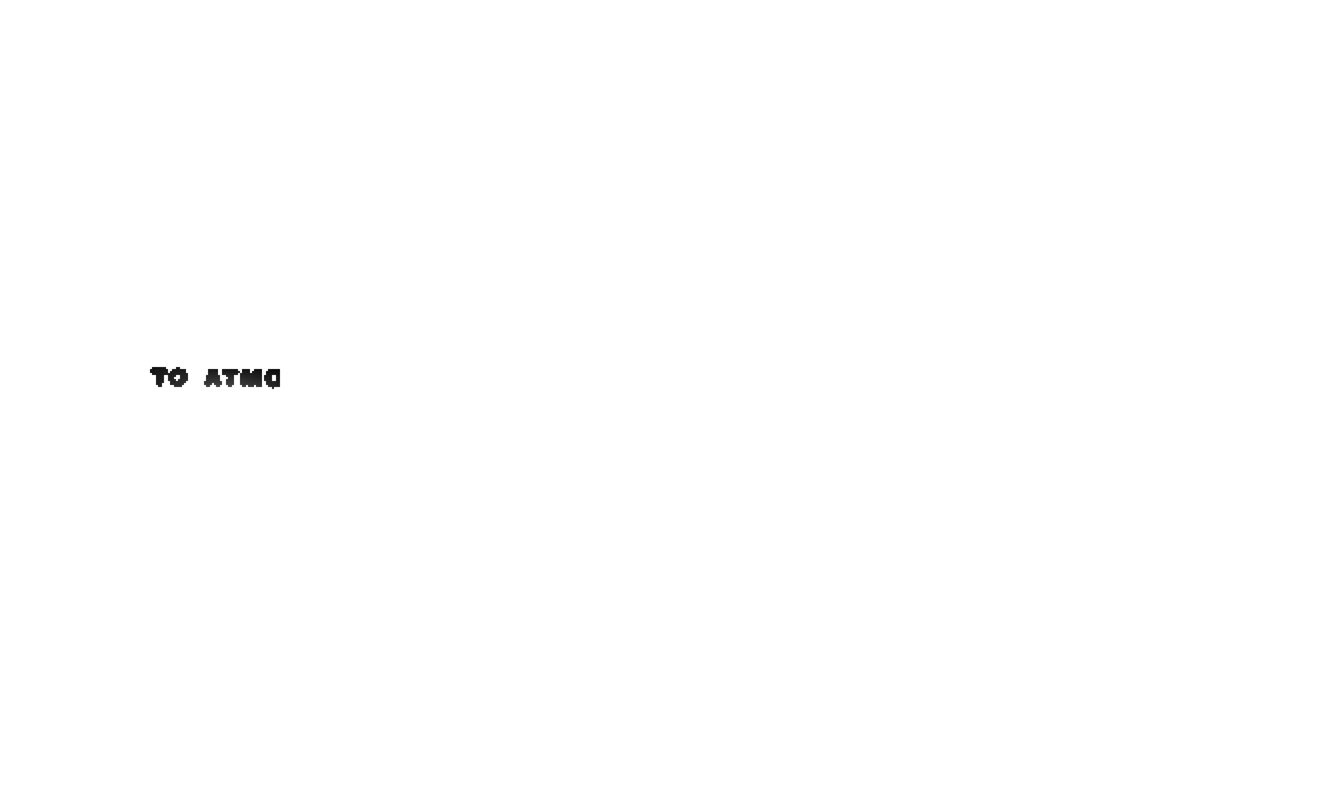


FIG. 252. Closed Heater Connected as an "Induced" Heater. Condensing Plant.

In the arrangement in Fig. 251 the number of fittings is reduced to a minimum and the heater may be readily cut out. Since induced heaters are apt to become air-bound, a vapor pipe or vent is inserted

in the top of the heater as shown. This pipe varies from ½ to 1½ inches in diameter, depending upon the size of heater.

Closed Heaters: Am. Elecn., May, 1900, p. 236, July, 1900, p. 354, Oct., 1905, p. 530; Cassier's Mag., Aug., 1903, p. 330; Eng. U.S., Jan. 1, 1906, p. 13; Power, April, 1902, p. 11.

263. Live-Steam Heaters and Purifiers. — The function of a live-steam heater and purifier is primarily that of purification and hence it is not ordinarily installed unless the feed water contains scale-forming elements such as sulphates of lime and magnesia. These, as previously stated, are not entirely precipitated until a temperature of approximately 300 degrees F. is reached; hence no amount of heating with exhaust steam at atmospheric pressure will thoroughly purify feed water containing these elements.

Fig. 253 shows a section through a Hoppes live-steam purifier. Since the purifier is subjected to full boiler pressure, the shell and

FIG. 253. Hoppes Live Steam Purifier.

heads are constructed of steel. Within the shell are a number of trough-shaped pans or trays placed one above another and supported on steel angle ways. Steam from the boiler enters the chamber at *A* and comes in contact with feed water and condenses. The water on entering the heater at *B* is fed into the top pan and, overflowing the edges, follows the under side of the pan to the center and drops into the pan below. It flows over each successive pan in the same manner until it reaches the chamber at the bottom, whence it gravitates to the boiler through pipe *C*. As the steam inclosed in the shell comes in contact with the thin film of water, the solids held in solution are separated and adhere to the bottom of the pans in the same manner

that stalactites form on the roofs of natural caves. Authentic tests show that live steam heaters do not increase the boiler efficiency and that they merely act as purifiers. (*Power*, March 31, 1908, p. 498.) The purifier should be set in such a position as will bring the bottom of the shell two feet or more above the water level of the boilers, as in Fig. 254. *N* is the feed pipe from pump to purifier and should be pro-

FIG. 254. Typical Installation of a "Live Steam" Purifier.

vided with a check valve. *D* is the gravity pipe through which the purified water flows to the boiler. This pipe should be carried below the water level of the boilers and all branch pipes should be taken off below the water line. Pipe *L* leads from top of pipe *S* to pump or other steam-using device. This is necessary in order that air and other non-condensable gases liberated from the water may be removed from the purifier, which would otherwise become air-bound. In the illustration the feed pump takes its supply from an exhaust steam heater *C*. The purifier is provided with a suitable by-pass so that the water may be fed directly to the boiler when necessary.

Live Steam Heated Feed Water: Elec. Engr., Lond., June 29, 1906; Am. Elecn., May, 1900, p. 214; Elec. Rev., Lond , May 20, 1898, p. 667; Eng. Rec., Aug. 30, 1898, p. 467; Power, March 31, 1908, p. 498.

264. Economizers. — Fig. 256 gives a general view of a Green economizer, illustrating a typical flue gas heater. It consists of a series of cast-iron tubes 9 to 12 feet in length and 4½-inches in diameter, which

FIG. 255 Typical Installation of Primary and Secondary Heaters.

FIG. 256. Green Economiser.

are arranged vertically in sections of various widths across the main flue between boiler and chimney. When in position the sections are connected by top and bottom headers, and the headers are connected to branch pipes running lengthwise, one at the top and the other at the bottom. Both of the branch pipes are outside the brickwork which incloses the apparatus. The waste gases are led to the economizer by the ordinary flue from the boiler to the chimney, but a by-pass must be provided for use when the economizer is out of service for cleaning or for repairs. The feed water is forced into the economizer through the lower branch pipe nearest the point of exit of gases, and emerges through the upper branch pipe nearest the point where the gases enter. Each tube is encircled with a set of triple overlapping scrapers which travel continuously up and down the tubes at a slow rate of speed, the object being to keep the external surfaces free from soot. The mechanism for working the scrapers is placed on top of the economizer, outside the chamber, and the motive power is supplied either by a belt from some convenient shaft or small independent engine or motor. The power for operating the gearing varies from 1 to $\frac{1}{2}$ horse power per 1000 square feet of economizer surface, depending upon the number and length of tubes. The apparatus is fitted with blow-off and safety valves, and a space is provided at the bottom of the chamber for the collection of soot.

Fig. 257 shows a typical installation of a fuel economizer.

FIG. 257. A Typical Economizer Installation. (*Eng. Rec.*)

265. Value of Economizers. — The general conclusion drawn from current practice is that an economizer installation results in —

A small annual saving in cost of operating the plant.

Decreased wear and tear on the boilers due to the higher feed-water temperature.

A large storage of hot water for sudden increase in load.

Purification of the feed water due to the high temperature in the economizer. The scale-forming elements do not bake hard on the economizer tubes as they would in the boiler where the heat from the fire is more intense, but make a muddy deposit readily removed by blowing off.

266. Factors Determining Installation of Economizers. — The factors to be considered before installing an economizer are:

The nature of the auxiliary machinery, direct connected or belted.

Method of heating the feed water; whether vacuum and atmospheric heaters are used and whether all or part of the auxiliary steam is used for heating.

Initial temperature of the feed water; whether the feed is taken from the hot well or from a cold supply.

Rise in temperature due to economizer.

Cost of economizer. An approximate price is $15 per tube erected, on a basis of 15 square feet per tube. The heating surface is rated at 3 to 5 square feet per boiler horse power.

Cost of additional building space.

Reduction in boiler-heating surface made possible by the economizer.

Extra cost of stack or forced-draft apparatus necessary to compensate for loss of draft due to economizer. The economizer lessens the draft by increasing the resistance between boilers and chimney and by reducing the chimney temperature. Where the installation of an economizer decreases the normal temperature of the chimney from say 550 degrees to 350 degrees F., the reduction in draft is approximately 25 per cent.

Total cost of economizer plant. This depends largely upon the design and varies from $4 to $7 per boiler horse power.

Interest, depreciation, repairs, operation, taxes, and insurance.

Table 67 gives the results of economizer tests.

267. Temperature due to Use of Economizer. — The rise in temperature of feed water due to the use of an economizer may be approximated from the following empirical formula advocated by the Green Economizer Company:

$$x = \frac{y\,(T_1 - t_1)}{9.1 + \frac{5\,w + GC}{2\,GC}\,y}, \tag{120}$$

in which

x = rise in temperature of the feed water.
T_1 = temperature of flue gas entering economizer.
t_1 = temperature of feed water entering economizer.
w = pounds of feed water per boiler horse power per hour.
G = pounds of flue gas per pound of combustible.
C = pounds of coal per boiler horse power per hour.
y = square feet of economizer heating surface per boiler horse power.

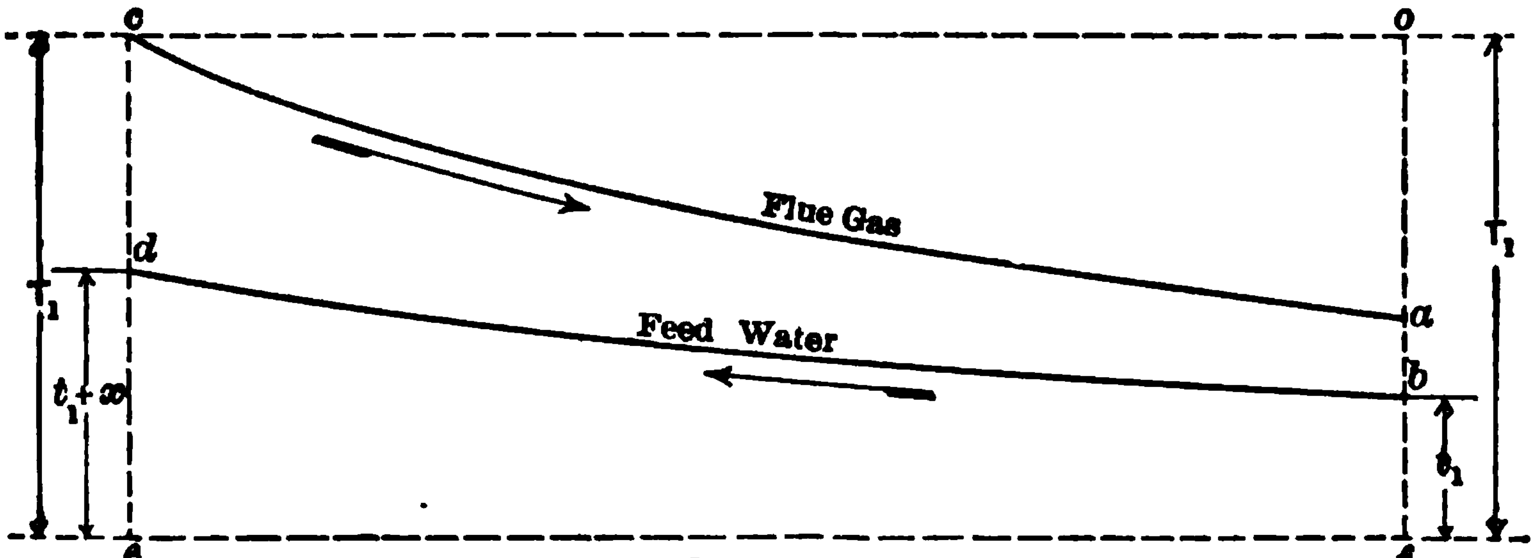


FIG. 258.

Referring to Fig. 258, the ordinates represent temperatures, and abscissas the path of the flue gas and the water in the economizer. The flue gas enters the economizer at c with temperature T_1 and leaves at a with temperature T. The feed water enters at b with temperature t_1 and leaves at d with temperature $t_1 + x$.

The algebraic mean temperature difference D between the flue gas and the feed water will be

$$D = \frac{cd + ab}{2} \tag{121}$$

$$= \frac{T_1 - (t_1 + x) + of - oa - bf}{2} \tag{122}$$

$$= \frac{T_1 - (t_1 + x) + T_1 - oa - t_1}{2} \tag{123}$$

$$= T_1 - t_1 - \frac{x + oa}{2}. \tag{124}$$

Now, wx = B.T.U. absorbed by the feed water per boiler horse power and

GCS = B.T.U. given up to the feed water by the flue gas for each degree reduction in temperature (S = specific heat of the flue gas); therefore

$wx \div GCS$ = total reduction in temperature of the flue gas; that is,

$$\frac{wx}{GCS} = oa. \tag{125}$$

Substituting (125) in (124), we get

$$D = T_1 - t_1 - \frac{x + \dfrac{wx}{GCS}}{2} \tag{126}$$

$$= T_1 - t_1 - x\frac{GCS + w}{2\,GCS}, \tag{127}$$

in which

D = mean temperature difference between flue gas and feed water, degrees F.

Let U = B.T.U. absorbed per hour per square foot per degree difference in temperature and

y = square feet of economizer surface per boiler horse power.

Then UDy = heat absorbed per boiler horse-power hour.

But wx = heat absorbed per boiler horse-power hour.

Therefore
$$wx = UDy. \tag{128}$$

Combining (128) and (127),

$$wx = Uy\left(T_1 - t_1 - \frac{x\,GCS + wx}{2\,GCS}\right), \tag{129}$$

from which
$$x = \frac{y\,(T_1 - t_1)}{\dfrac{w}{U} + \dfrac{w + GCS}{2\,GCS}}. \tag{130}$$

y varies from 3.5 square feet to 5 square feet per boiler horse power, and U from 2.25 to 3.3, depending upon the conditions of operation.*

If we let $w = 30$, $S = 0.2$, and $U = 3.3$, and substitute these values in equation (130), it assumes the form given by the Green Economizer Company, equation (120).

A method of approximating the rise in temperature where the final temperature of the flue gas is known, is to assume $\frac{5}{10}$ degree rise in the feed water for each degree reduction in temperature in the flue gas. This is determined on the basis that approximately 20 pounds of flue gas are generated for each pound of combustible, and that 10 pounds of water are evaporated per pound of combustible; that is, 2 pounds of flue gas are generated for each pound of feed water delivered to the boiler. Assuming a specific heat of 0.25 for the flue gas, this gives $2 \times .25$ or 0.5 degree rise in temperature in the feed water for each degree reduction in the flue gas temperature.

Example: Determine the rise in temperature of the feed water in a power plant of 1200 I.H.P. Engines use 20 pounds of steam per

* For $D = 600$ $U = 3.25$ For $D = 400$ $U = 2.75$
500 3.00 300 2.25

I.H.P. hour; auxiliaries use the equivalent of 12 per cent of main engine steam; vacuum 25 inches; feed-water supply 50 degrees; 3.7 pounds of coal are burned per hour per boiler horse power; flue gas temperature 550 degrees F.; steam pressure 150 pounds gauge.

The vacuum and atmospheric heater will raise the temperature of the feed water from 50 to 205 degrees. (See preceding problem.)

On the assumption that 20 pounds of flue gas are generated per pound of combustible and that 3.5 square feet of economizer heating surface are installed per boiler horse power, the notations in the formula will become $T_1 = 550$, $t_1 = 205$, $w = 30$, $G = 20$, $C = 3.7$, $y = 3.5$, $U = 3.3$, $S = 0.2$.

Substituting,

$$x = \frac{3.5\,(550 - 205)}{\dfrac{30}{3.3} + \dfrac{(5 \times 30 + 20 \times 3.7)}{(2 \times 20 \times 3.7)}\,3.5}.$$

$= 83$ degrees rise in temperature.

Therefore the temperature of the water entering the boiler will be

$205 + 83 = 288$ degrees F.

Economizers: Prac. Engr. U.S., March, 1910; Engr. U.S., March 15, 1900, p. 69; Power, July 27, 1909; Cassier's, March, 1900, p. 378; St. Ry. Jour., Oct. 31, 1903, 1903, p. 822.

Economizer Installations: Eng. Rec., March 15, 1902, p. 247, June 7, 1902, p. 532, Nov. 1, 1902, p. 410; Power, Feb., 1905.

Tests of Economizers: Am. Elecn., Nov., 1902, p. 518; Elec. Rev., Lond., Aug. 2, 1895, p. 148; Eng. Rec., March 9, 1901, p. 220, July 25, 1903, p. 102; Power, Aug., 1897, p. 8, Aug., 1904, p. 493; St. Ry. Rev., June, 1904, p. 436.

TABLE 67.

ECONOMIZER PERFORMANCES.*

Plants Tested.	Temperatures.					Fuel Saving.
	Gases Entering Economizer.	Gases Leaving Economizer.	Water Entering Economizer.	Water Leaving Economizer.	Gain in Temperature of Water.	
	Degrees.	Degrees.	Degrees.	Degrees.	Degrees.	Per Cent.
1	610	340	110	287	177	16.7
2	505	212	84	276	192	17.1
3	550	205	185	305	120	11.7
4	522	320	155	300	145	13.8
5	505	320	190	300	110	10.7
6	465	250	180	295	115	11.2
7	490	290	165	280	115	11.0
8	495	190	155	320	165	15.5
9	595	299	130	311	181	16.8

* Transactions American Society of Mechanical Engineers. Vol. XV.

268. Choice of Feed-Water Heating System. — The heating of feed water and its delivery to the boiler in the most economical manner is a problem involving such a large number of combinations that a general analysis is impracticable. The following discussion of a specific case will give some idea of the manner in which this problem may be attacked.

Example: Determine the most economical manner of heating the feed water for a power plant of 1000 horse power operating under the following conditions: Schedule 10 hours per day and 310 days per year; load factor on the ten-hour basis 0.8; cost of coal $2.50 per ton of 2000 pounds; heat value of the coal 13,500 B.T.U. per pound; average boiler efficiency 65 per cent; engines use 20 pounds of steam per I.H.P. hour; steam pressure 150 pounds absolute; temperature of cold water 60 degrees; vacuum 26 inches referred to 30-inch barometer; interest 5 per cent; depreciation 8½ per cent; maintenance 1 per cent; insurance ½ per cent; taxes 1 per cent; total charges 16 per cent; charges for attendance and maintenance assumed to be the same in each case and credit for the chimney assumed to offset debit for economizer space. Many of the influencing conditions are left out for the sake of simplicity.

The most likely combinations are

(1) Atmospheric, all auxiliaries steam driven, water taken from cold well.

(2) Same as 1 except that water is taken from hot well.

(3) Economizers, auxiliaries electrically driven, chimney draft, water from cold well.

(4) Vacuum heater, economizer, and electrically driven auxiliaries, fan draft.

(5) Vacuum heater, atmospheric heater, and steam auxiliaries.

(6) Atmospheric heater, economizer, steam auxiliaries, fan draft.

(7) Vacuum and atmospheric heaters, economizers, steam auxiliaries, and electrical fan.

(8) Vacuum, atmospheric heater, economizer, and chimney draft, auxiliaries operating condensing except feed pumps and stoker engines which exhaust into the atmospheric heater.

The difference between the total heat furnished by the boiler and the heat returned in the feed water is the net heat put into the steam by the boiler. Evidently the system which shows the least net heat required to produce one horse power will be the most economical as

far as coal consumption is concerned, although not necessarily the cheapest when both operating and fixed charges are considered.

Prices vary so much that it is practically impossible to give costs of installations which will bear criticism and the prices taken in this problem are approximate only.

CASE I.

Atmospheric heater, auxiliaries steam driven, feed from cold well.

This arrangement and that of Case II are the most common in power plants of this size.

The power consumption of the auxiliaries operating non-condensing varies from 8 to 12 per cent of the total power developed. Assume it to be 10 per cent.

The temperature of the feed water leaving the heater may be determined by formula (105).

$$t = \frac{t_0 + .9\,S\,(\lambda + 32)}{1 + .9\,S}.$$

Substituting $S = .10$, $\lambda = 1146$, $t_0 = 60$,

$$t = \frac{60 + .9 \times .10\,(1146 + 32)}{1 + .9 \times .10}$$
$$= 152.$$

The *net heat* furnished by the boiler to produce one indicated horse-power hour in the engine is evidently the heat necessary to raise 20 + 10 per cent of 20 = 22 pounds of water from 152 degrees F. to steam at 150 pounds pressure; i.e., the net heat furnished is

$$22 \times 1071.2 = 23{,}564 \text{ B.T.U.}$$

Now, 1 I.H.P. = 2545 B.T.U.

Therefore the heat efficiency of this arrangement is

$$\frac{2545}{23{,}564} = 10.8 \text{ per cent.}$$

Probable First Cost.

Steam pumps	$400.00
Condenser with steam-driven air and circulating pumps	3000.00
1000 horse-power open heater	480.00
Piping	1200.00
	$5080.00

Fuel Consumption.

Average horse-power hours per year = 1000 (rated horse power) $\times$ 0.8 (curve load factor) $\times$ 310 (days per year) $\times$ 10 (hours per day) = 2,480,000.

Pounds of coal per I.H.P. hour = net heat furnished per I.H.P. hour $\div$ net heat absorbed by the boiler per pound of coal = 23,564 $\div$ (13,500 $\times$ 0.65) = 2.68.

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.68}{2000} = 3323.$$

Fuel and Fixed Charges.

Fuel, 3323 tons at $2.50	$8,308.00
Fixed charges, 16 per cent of $5080	812.00
	$9,120.00

CASE II.

Same as Case I, except that feed is taken from the hot well. This arrangement is possible only when the condensing water is suitable for feed purposes.

Assume the temperature of the water from the hot well as it enters the heater to be 110 degrees.

The temperature of the feed water leaving the heater will then be 198 degrees (from formula (105)).

$$\text{Net heat furnished} = 22 \times 1025.2 = 22{,}554 \text{ B.T.U.}$$

$$\text{Efficiency} = \frac{2545}{22{,}554} = 11.3 \text{ per cent.}$$

$$\text{Pounds of coal per I.H.P.} = \frac{22{,}554}{13{,}500 \times .65} = 2.62.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.62}{2000} = 3248.$$

Fuel and Fixed Charges.

Fuel, 3248 tons at $2.50	$8,120.00
Fixed charges (same as Case I)	812.00
	$8,932.00

CASE III.

Economizers, auxiliaries electrically driven, chimney draft, water from the cold well.

Practice gives an average of 3 per cent of the main engine output as the power required to operate the electrical auxiliaries in a plant of this size.

The temperature of the feed water leaving the economizer may be determined from formula (120).

$$x = \frac{y\,(T_1 - t_1)}{9.1 + \frac{5\,w + GC}{2\,GC}\,y}.$$

Substituting,

$$x = \frac{3.5\,(550 - 60)}{9.1 + \frac{5 \times 30 + 20 \times 3.7}{2 \times 20 \times 3.7}\,3.5} = 119 \text{ degrees.}$$

Temperature of feed water entering heater = 119 + 60 = 179 degrees.
Net heat furnished = (20 + 3 per cent of 20) × 1044.2 = 21,510 B.T.U.

$$\text{Efficiency} = \frac{2545}{21{,}510} = 11.8 \text{ per cent.}$$

Probable First Cost.

Economizers	$3,500.00
Motor feed pump	600.00
Condenser with electrically driven air and circulating pump	6,000.00
Piping and wiring	1,000.00
	$11,100.00

Fuel Consumption.

$$\text{Pounds of coal per I.H.P. hour} = \frac{21{,}510}{13{,}500 \times .65} = 2.45.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.45}{2000} = 3038.$$

Fuel and Fixed Charges.

Fuel, 3038 tons at $2.50	$7,595.00
Fixed charges, 16 per cent on $11,100	1,776.00
	$9,371.00

CASE IV.

Vacuum heater, economizer, electrically driven auxiliaries, fan draft.

The vacuum heater may be relied upon to raise the temperature of the feed water to 110 degrees.

The economizer will increase this 107 degrees (from formula (120)), giving the feed water a temperature of 217 degrees as it enters the heater.

The electrical fan for the mechanical-draft system will require approximately 2 per cent of the main system engine power, making a total of 3 + 2 = 5 per cent for all auxiliaries.

$$\text{Net heat furnished} = (20 + 5 \text{ per cent of } 20) \times 1006.2.$$
$$= 21{,}130 \text{ B.T.U.}$$
$$\text{Efficiency} = \frac{2545}{21{,}130} = 12.05 \text{ per cent.}$$

Probable First Cost.

For the sake of simplicity it is assumed that the high first cost of the chimney plus its low depreciation and maintenance will offset the low first cost of the mechanical-draft system plus its higher maintenance and depreciation charges.

Economizers	$3,500.00
Motor feed pump	600.00
Motor-driven pumps and condenser	6,000.00
Motor-driven fan	750.00
Piping and wiring	1,200.00
Vacuum heater	200.00
	$12,250.00

Fuel Consumption.

$$\text{Pounds of coal per I.H.P. hour} = \frac{21{,}130}{13{,}500 \times .65} = 2.41.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.41}{2000} = 2988.$$

Fuel and Fixed Charges.

Fuel, 2988 tons at $2.50	$7,470.00
Fixed charges, 16 per cent of $12,250	1,960.00
	$9,430.00

In like manner Cases V, VI, VII and VIII have been treated and are tabulated in the summaries.

SUMMARY (1).

Case.	Temperature of Feed Water.	Power Consumed by Auxiliaries.	Efficiency.	First Cost.	Fuel Cost per Year.	Cost of Operation per Year.
	Degrees F.	Per Cent.	Per Cent.			
I	152	10	10.8	$5,080	$8,308	$9,120
II	198	10	11.3	5,080	8,120	8,932
III	179	3	11.8	11,100	7,595	9,371
IV	217	5	12.05	12,250	7,470	9,430
V	208	10	11.4	5,280	7,900	8,744
VI	294	14	12	9,000	7,750	9,190
VII	290	10	12.2	9,300	7,380	9,570
VIII	270	8	12.3	8,250	7,075	8,395

SUMMARY (2).

Case.	Efficiency.	First Cost.	Fuel.	Cost per Year.
I	8	1	8	4
II	7	1	7	2
III	6	6	4	6
IV	3	7	3	7
V	5	2	6	3
VI	4	4	5	5
VII	2	5	2	8
VIII	1	3	1	1

Summary (2) gives the ranking; thus: Case I is eighth in point of efficiency; first in cheapness of installation; eighth in yearly cost of fuel; and fourth in yearly cost of operation. Case VIII is apparently the best arrangement for the *given conditions.*

CHAPTER XIII.

PUMPS.

269. Classification. — Pumps used in connection with steam power plants may be conveniently classified under five groups according to the principles of action.

1. *Piston pumps,* in which motion and pressure are imparted to the fluid by a reciprocating piston, plunger, or bucket. The action is positive and a certain definite amount of fluid is handled per stroke under predetermined conditions of pressure and velocity.

2. *Centrifugal pumps,* in which the fluid is given initial velocity and pressure by a rotating impeller. The action is not positive, as the amount of fluid discharged is not necessarily proportional to the impeller displacement.

3. *Rotary pumps,* in which motion and pressure are imparted to the fluid by a rotating impeller. The volume discharged is practically equal to the impeller displacement regardless of pressure.

4. *Jet pumps,* in which velocity and pressure are imparted to the fluid by the momentum of a jet of similar or other fluid. The ordinary steam injector is the best known of this group.

5. *Direct-pressure pumps,* in which the pressure of one fluid acts directly on the surface of another fluid, thereby imparting all or part of its energy to the latter. The pulsometer is an example of this type.

These groups may be variously subdivided as follows:

Piston	Direct-acting	Simplex, Duplex	Air. Vacuum. Forcing. Lifting.
	Fly-wheel, Power driven	Simplex, Duplex, Triplex	
Centrifugal	Volute	Single stage	Forcing. Lifting.
	Turbine	Multi-stage	
Rotary	Power driven	Forcing, Lifting	
Jet	Injector	Positive, Automatic	
Direct pressure	Pulsometer	Lifting	
	Air-lift	Lifting	

Piston or *plunger* pumps are the most common in use. Boiler-feed pumps, city waterworks pumps, and force pumps are ordinarily of this type. In the direct-acting type, Fig. 260, the water

plunger and steam piston are secured to a single piston rod and the steam pressure is transmitted directly to the water. There is no fly wheel, connecting rod, or crank. The velocity of the delivery is proportional to the resistance offered by the water; when the resistance equals the forward effort of the steam pressure the pump stops. This class of pump is well adapted for boiler-feeding purposes, since it may be operated as slowly as suits the requirements of feeding by simply throttling the discharge. The steam consumption is very large in proportion to the work performed, since the steam is not used expansively.

Fly-wheel pumps, Figs. 273, 308, are ordinarily classified as pumping engines. In this class steam may be used expansively, as sufficient energy is stored in a fly wheel to permit the drop in steam pressure during expansion. These pumps find wide application in city waterworks, elevator plants, and the like, where high duty is required. They are little used as stationary boiler feeders, but are used to some extent in river boat practice and in plants operating continuously for long periods at comparatively steady loads. Practically all sizes of dry-air pumps and a number of large jet condenser pumps are of this type.

Piston pumps, Fig. 279, driven by gearing or belting are ordinarily classified as power-driven pumps. The driving power may be steam engine, electric motor, or gas engine. The single-cylinder machine is often designated as a "simplex" power-driven pump, the two-cylinder as a "duplex," the three-cylinder as a "triplex," and so on.

Centrifugal pumps, Fig. 292, are supplanting to a considerable extent the present type of piston pump for many uses. Though particularly adapted for low heads and large volumes they are used in many situations requiring extremely high heads. They are not as efficient as high-grade pumping engines, but the extremely low first cost frequently offsets this disadvantage, and they are much used in connection with dry docks, irrigating plants, sewage systems, and as circulating pumps in condensing plants.

Rotary pumps, Fig. 305, are employed to a limited extent in the same field as the centrifugal pump. Being positive in action, they permit of a much lower rotative speed for the same delivery pressure.

Jet pumps, Fig. 282, are seldom used as pumps in the ordinary sense of the word, on account of their extremely low efficiency, but are frequently employed for discharging water from sumps. Their greatest field of application lies in boiler feeding and in this respect their efficiency is comparable with that of the average piston pump.

During this time the opposite piston has completed a full stroke and the valve operated by it will have opened the steam port wide, so that while one valve covers both steam ports the other is at the end of its travel. In some makes of pumps the stem is rigidly attached to the valves, the lost motion being adjusted outside the steam chest as shown in Figs. 261 and 262, which represent two common constructions of duplex valve gear.

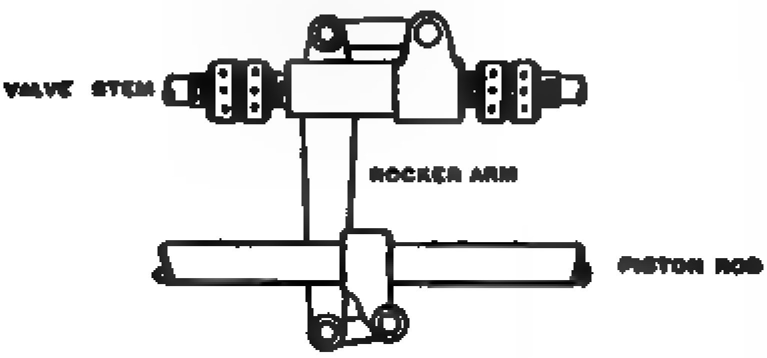


FIG. 262.

Fig. 263 shows the valve and piston in the position occupied at the commencement of the stroke. At one end of the valve the steam port P is open wide and at the opposite end the exhaust port E is open wide. When the piston nears the opposite end of the stroke and reaches the position shown in Fig. 264 the steam escape through the exhaust port E is cut off by the piston, and since the steam port is closed, the remaining steam is compressed between the piston and cylinder head, thus arresting the motion of the piston gradually without shock or jar.

FIG. 263.

The construction of the water end of single-cylinder and duplex pumps is practically the same; any slight differences which may be found are confined to minor details which in no way affect the general design or operation of the pump. The piston is double acting, the single-acting cylinder being confined to power pumps or to steam pumps intended for very high pressures. In the old-style pumps it was the custom to use one large valve with a lift sufficient to give the required passage, but in modern practice the required area is divided among several small

FIG. 264.

valves, so that each one is easily and cheaply removed in case of accident or wear, and slip is lessened.*

The valves are carried by two plates or decks, the suction valves being attached to the lower plate and the delivery valves to the upper one, as shown in Fig. 260.

The valves in practically all boiler-feed pumps are of the flat disk type, Fig. 265, held firmly to the seat by conical springs and guided by a bolt through the center.

All pumps are provided with an air chamber on the discharge side, which acts as a cushion for the water, prevents excessive pounding, and insures a uniform flow. Fig. 266 shows a section through the steam end of a compound duplex pump.

FIG. 265. A Typical Pump Disk-Valve.

FIG 266. Section Through Steam Cylinders of a Typical Compound Duplex Pump.

271. Feed Pumps with Steam-Actuated Valves. — Single-cylinder direct-acting pumps, Fig. 267, are ordinarily operated by steam-actuated valves. The steam enters the chest *C* and passes to the left through the annular opening *A* formed between the reduced neck of the valve and the bore of the steam chest. It is thus projected against the inside surface of the valve head *H* before escaping through the port *P* and passing to the cylinder. Both the pressure and impulse due to velocity acting on the valve head *H* tend to close or restrict the

* The modern Riedler pump is an exception. See *Engineer, U.S.*, Nov. 15, 1907, p. 1040.

admission port by forcing the valve to the left. On reaching the cylinder and forcing the piston X toward the right, the pressure of the steam upon the opposite side of the valve head H is pressing the valve to the right, a movement which would give the admission more port opening at A and deliver more steam to the cylinder. The valve then holds a position depending upon the relative intensity of the two pressures, which tend to move it in opposite directions, the admission steam, tending to close the valve, and cylinder steam, tending

FIG. 267. Marsh Boiler-Feed Pump. A Typical Steam-Actuated Valve Gear.

to open the valve wider. The steam valve, therefore, is always in a balanced position. The steam piston is grooved at the center, forming a reservoir for live steam R which is supplied from the upper chamber of the steam chest by passage E to the cylinder cap S, and thence by tube M and the hollow piston rod V. The steam in this annular piston space reverses the steam valve by pressing alternately against the outer surfaces of the valve heads H through the connecting passages O, O near each end of the cylinder. The tappets T are for the purpose of moving the valve by hand in case it fails to

move automatically. Steam-actuated valves are not as positive in action as mechanically operated valves, and hence are little used in situations where positive action is essential, as in fire-pump service.

272. Air and Vacuum Chambers. — Air chambers in piston pumps are for the purpose of causing a steady discharge of water and of reducing excessive pounding at high speeds by providing a cushion for the water. The water discharged under pressure compresses the air in the air chamber somewhat above the normal pressure of discharge during each stroke of the water piston, and when the piston stops momentarily at the end of the stroke the air expands to a certain extent and tends to produce a uniform rate of flow.

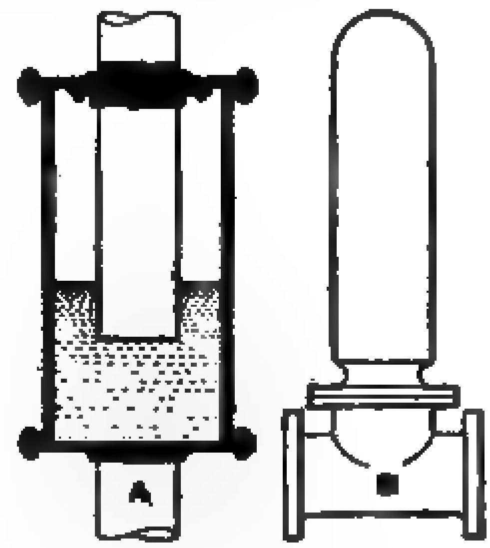


Fig. 268. Forms of Vacuum Chambers.

The volume of the air chamber varies from 2 to $3\frac{1}{2}$ times the volume of the water piston displacement in single-cylinder pumps, and from 1 to $2\frac{1}{2}$ times in the duplex type. High-speed pumps are provided with air chambers of from 5 to 6 times the piston displacement. The water level in the air chamber should be kept down to one-fourth the height of the chamber. In slow-running pumps sufficient air may be carried into the pump chamber along with the water, but with high speeds a large part of the air will be discharged, and air must be forced into the chamber by mechanical means. The larger the chamber the more uniform will be the discharge pressure.

(A) (C)

Fig. 269. Different Arrangements of Vacuum Chambers.

Vacuum chambers are frequently provided for the purpose of maintaining a uniform flow of water in the suction pipe and assisting in the reduction of slip. Such chambers should be of slightly greater volume

than the suction pipe and of considerable length rather than diameter. Fig. 268 illustrates two designs commonly used. The one in Fig. 268 (B) should be placed in such a position as to receive the impact of the column of water in the suction pipe as illustrated in Figs. 269 (A), 269 (B), and 269 (C). The chamber illustrated in Fig. 268 (A) should be placed in the suction pipe below but close to the pump.

273. Water Pistons and Plungers. — In cold-water pumps the water pistons are usually packed with some kind of soft packing. Fig. 270 (A)

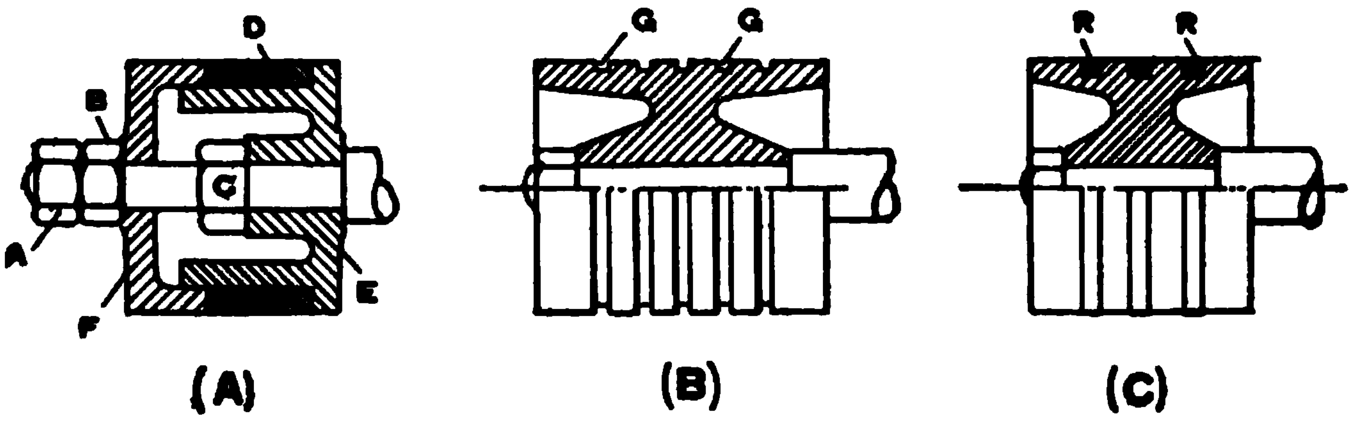


FIG. 270. Types of Water Pistons.

shows the details of a piston with square *hydraulic packing*. The body E is fastened to the piston rod by nut C; packing is placed at D, and follower F is forced up by the nut B and locked by nut A. For large

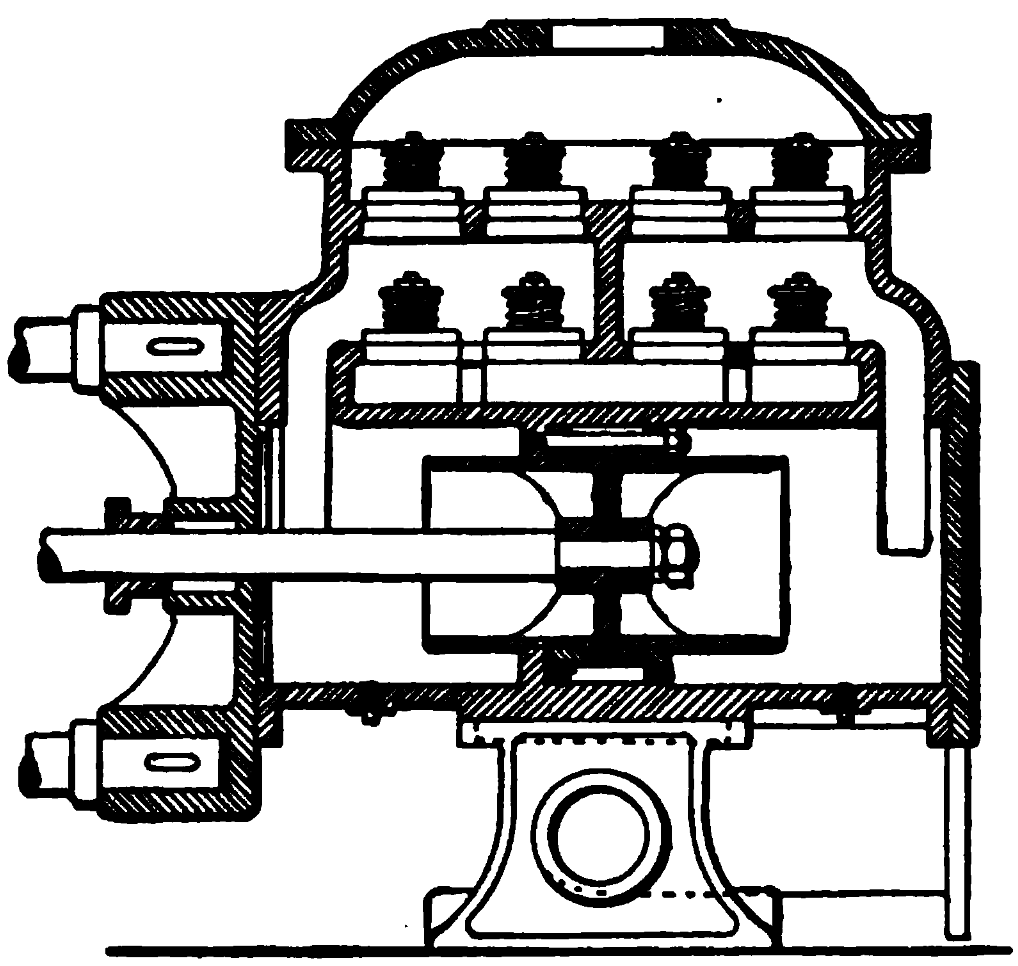


FIG. 271. Plunger with Metal Packing Ring.

sizes the design is the same except that the follower is set up by a number of nuts near the edge. In hot-water pumps the pistons are often packed by means of *metallic piston rings* R, R, Fig. 270 (C), similar to those in steam pistons, or merely by *water grooves* G, G, Fig. 270 (B).

The water end is often fitted with a *plunger* instead of a piston, as in Figs. 271 to 273. The piston is more compact, but the plungers do not require a bored cylinder, so that the first cost is not materially different.

Fig. 271 shows a plunger with metal packing ring. When leakage becomes excessive it is necessary to renew the ring, which is readily removed.

In Fig. 272 the plunger is packed with hydraulic packing as in the follower type of pump piston. The great difficulty with the above

FIG. 272. Plunger with Hydraulic Packing.

types of piston and plunger is in keeping the packing tight or in knowing when it is leaking, and the trouble necessary to replace the packing. The *outside packed plunger*, Fig. 273, obviates these disadvantages to a great extent, since leakage is readily detected and repacking is performed without removing the cylinder heads. In dirty, dusty locations, however, the piston pump or inside packed plunger is to be preferred, since the abrasive action of the dust renders outside packing difficult. Fig. 273 illustrates a high-duty elevator pump with outside packed plunger.

274. Performance of Piston Pumps. — Direct-acting pumps as a class are wasteful of fuel and low in efficiency, due largely to the non-expansive use of steam. The average small duplex boiler-feed pump uses from 100 to 200 pounds of steam per I.H.P. hour, depending upon the speed, and the mechanical efficiency varies from 50 per cent

Fig. 273. Horizontal Fly-Wheel Pump with Outside Packed Plunger.

to 90 per cent. When new and in proper working condition the mechanical efficiency is seldom less than 85 per cent; but such pumps, as a rule, are given scant attention, and the average efficiency is not far from 65 per cent. The term "mechanical efficiency" in this connection refers to the ratio of the actual water horse power to the indicated horse power of the steam cylinder. The loss includes the slip of the piston and valves. A steam consumption of 150 pounds per I.H.P. hour with mechanical efficiency of 65 per cent is equivalent to a power consumption of about 5 per cent of the rated boiler capacity, although if the exhaust steam is used for feed-water heating the actual heat consumption may be but 1 to 1.5 per cent. Compound direct-acting pumps running non-condensing use from 50 to 100 pounds of steam per I.H.P. hour. Single-cylinder fly-wheel pumps of the slow-speed type, running non-condensing,. use about 50 pounds of steam per I.H.P. hour. Multi-cylinder fly-wheel pumps of the high-duty type use about 25 pounds per I.H.P. hour when running non-condensing, and as low as 10 pounds when operating condensing. High-grade *direct-connected* motor-driven power pumps have a mechanical efficiency from line to water load, at normal rating, of about 80 per cent. The efficiency of *geared* pumps at normal rating varies with the character of the gearing and the degree of speed reduction, and may range anywhere from 40 to 70 per cent.

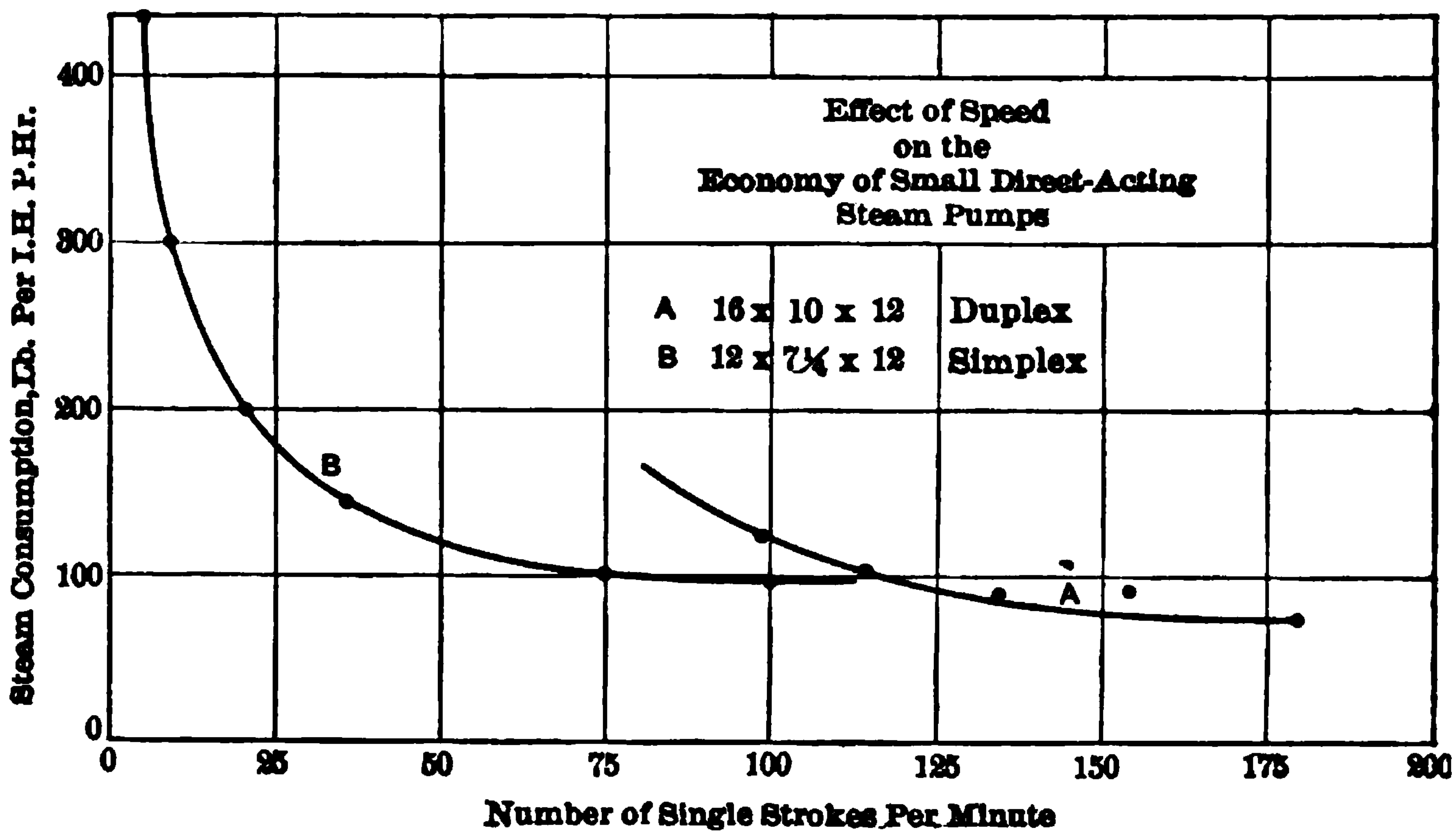


FIG. 274.

The steam consumption of all direct-acting boiler pumps decreases with the increase in speed. This is illustrated by curve *B*, Fig. 274, plotted from the tests of a 12 x 7¼ x 12 direct-acting single-cylinder pump at Armour Institute of Technology, and curve *A* based on experi-

ments with a 16 x 12 duplex fire pump at Massachusetts Institute of Technology.

Fig. 275 gives the details of the performance of a 12 x 7½ x 12 Marsh boiler-feed pump at the Armour Institute of Technology.

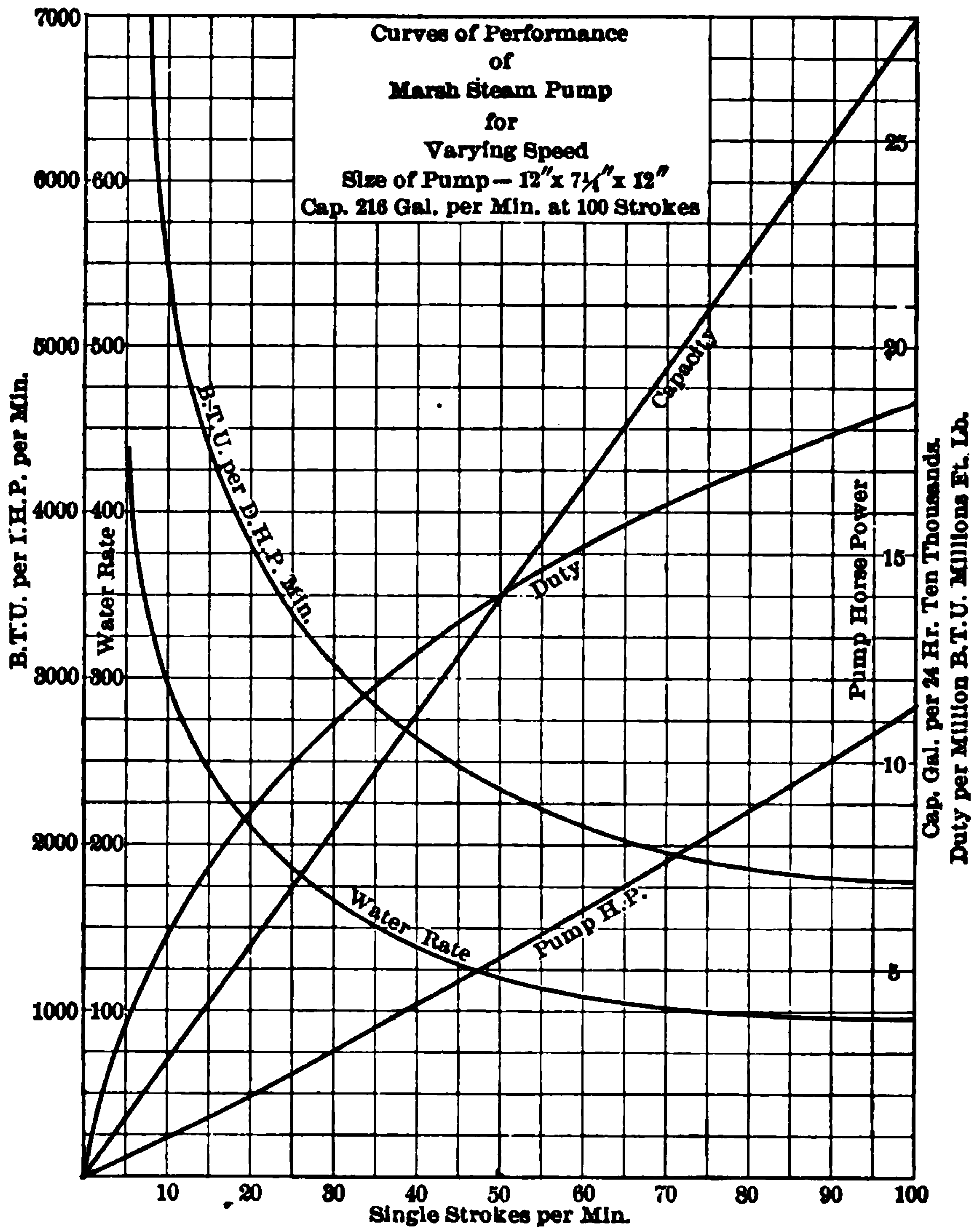


Fig. 275.

The determination of the power consumption of a boiler-feed pump is best illustrated by the following example.

Example: A small direct-acting duplex pump uses 150 pounds of steam per I.H.P. hour. Gauge pressure 150 pounds per square inch; feed-water temperature 64 degrees F. Required the per cent of rated boiler capacity necessary to operate the pump.

The head pumped against, 150 pounds per square inch, is equivalent to 150 $\times$ 2.3 = 345 feet of water.

The friction through the valves, fittings, and pipe, and the vertical distance between suction and feed-water inlet, are assumed to be equivalent to 20 per cent of the boiler pressure, giving a total head of 150 + 30 = 180 pounds per square inch, or 414 feet of water.

A boiler horse power, taking into consideration leakage losses and the steam used by the feed pump, will be equivalent to the evaporation of approximately 32 pounds of water per hour from a feed temperature of 64 degrees F. to steam at 150 pounds gauge.

The actual work done in pumping 32 pounds of water against a head of 414 feet is

$$414 \times 32 = 13{,}248 \text{ foot-pounds.}$$

This corresponds to

$$\frac{13{,}248}{60 \times 33{,}000} = 0.0067 \text{ horse power.}$$

The total heat of one pound of steam above 64 degrees F. is 1161.2 B.T.U. The heat delivered to the pump per I.H.P. hour is

$$1161.2 \times 150 = 174{,}180 \text{ B.T.U.}$$

The amount used by the pump for each boiler horse power, disregarding efficiency, is

$$174{,}180 \times 0.0067 = 1167 \text{ B.T.U. per hour.}$$

The mechanical efficiency of the average feed pump ranges from 50 to 85 per cent, depending upon its condition and the number of strokes per minute. Assuming it to be 65 per cent, the heat used by the pump per hour to deliver 32 pounds of water into the boiler is

$$1167 \div 0.65 = 1795 \text{ B.T.U.}$$

A boiler horse power is equivalent to 33,320 B.T.U. per hour. Therefore the per cent of boiler output necessary to operate the pump is

$$100 \times \frac{1795}{33{,}320} = 5.4 \text{ per cent.}$$

If the exhaust steam is used for heating the feed water, the steam consumption will be 1.37 per cent of the boiler capacity, thus: The weight of steam consumed per boiler horse-power hour

$$\frac{1795}{1161.2} = 1.54 \text{ pounds.}$$

Allowing 10 per cent for condensation, the heat in the exhaust available for heating the feed water is

$$966 \times 0.90 \times 1.54 = 1340 \text{ B.T.U.*}$$

* Surface Condenser Plant.

$1795 - 1340 = 455$ B.T.U., or the net heat required by the pump per hour to deliver 32 pounds of water to the boiler.

The per cent of boiler output necessary to operate the pump is

$$100 \frac{455}{33,320} = 1.37.$$

Pump performances are generally given in terms of the foot-pounds of work done by the water piston per thousand pounds of dry steam or per million B.T.U. consumed by the engine, thus:

$$1. \quad \text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Weight of dry steam used}} \times 1000.$$

$$2. \quad \text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat units consumed}} \times 1,000,000.$$

(See A.S.M.E. code for conducting duty trials of pumping engines, Trans. A.S.M.E., 12–530, 563.)

Example: A compound feed pump uses 100 pounds of steam per I.H.P. hour; indicated horse power, 48; capacity, 400 gallons per minute; temperature of water, 200 degrees F.; total head pumped against, 175 pounds per square inch; steam pressure, 100 pounds gauge; moisture in the steam, 3 per cent. Required the duty on the dry steam and on the heat-unit basis.

175 pounds per square inch is equivalent to $175 \times 2.4 = 420$ feet of water at 200 degrees F.

Weight of 400 gallons of water at 200 degrees F. $= 400 \times 8.03 = 3212$ pounds.

Work done per minute $= 3212 \times 420 = 1,349,040$ foot-pounds.

Weight of dry steam supplied per minute $= \frac{100 \times 48}{60} \times 0.97 = 77.6$ pounds.

B.T.U. supplied per minute $= \frac{100 \times 48}{60} (0.97 \times 876.2 + 308.8 - 200 + 32) = 79,256.$

Duty per thousand pounds of dry steam

$$= \frac{1,349,040}{77.6} \times 1000 = 17,384,150 \text{ foot-pounds.}$$

Duty per million B.T.U.

$$= \frac{1,349,040}{79,256} \times 1,000,000 = 16,893,863 \text{ foot-pounds.}$$

Table 68 may be used in approximating the duty, thus:

The mechanical efficiency of the pump in the preceding problem is

$$\text{Efficiency} = \frac{\text{P.H.P.}}{\text{I.H.P.}} = \frac{1,349,040}{33,000 \times 48} = 85 \text{ per cent.}$$

TABLE 68.

PERFORMANCE OF STEAM PUMPS. DUTY IN MILLIONS OF FOOT-POUNDS PER MILLION B.T.U.

Mechanical efficiency $= \dfrac{\text{Pounds discharged per min.} \times \text{head in feet}}{\text{I.H.P. of steam cylinder} \times 33{,}000}$.

Steam Consumption, Lbs. per I.H.P. Hour.		0.95	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55	0.50
Initial Press. 100 Lbs. Gauge, Non-Cond.	200	9.40	8.91	8.42	7.92	7.42	6.93	6.44	5.94	5.45	4.95
	190	9.90	9.39	8.86	8.36	7.83	7.31	6.79	6.27	5.74	5.22
	180	10.45	9.90	9.38	8.80	8.25	7.70	7.15	6.60	6.05	5.50
	170	10.90	10.50	9.90	9.32	8.74	8.15	7.57	6.99	6.41	5.83
	160	11.75	11.13	10.51	9.90	9.28	8.66	8.04	7.42	6.81	6.19
	150	12.55	11.90	11.22	10.59	9.90	9.29	8.49	7.92	7.26	6.60
	140	13.45	12.75	12.02	11.32	10.61	9.90	9.20	8.49	7.76	7.07
	130	14.49	13.71	12.96	12.20	11.42	10.69	9.90	9.15	8.38	7.62
	120	15.67	14.85	14.03	13.20	12.38	11.55	10.71	9.90	9.08	8.25
	110	17.10	16.21	15.31	14.40	13.50	12.60	11.70	10.80	9.90	9.00
	100	18.81	17.82	16.82	15.82	14.84	13.86	12.88	11.88	10.89	9.90
	90	20.90	19.80	17.76	16.72	15.67	14.63	13.58	12.54	11.49	10.45
	80	23.51	22.27	21.03	19.80	18.56	17.32	16.08	14.85	13.61	12.37
	70	26.90	24.50	24.04	22.64	21.22	19.80	18.40	16.98	15.58	14.14
	60	31.35	29.70	28.05	26.40	24.75	23.10	21.43	19.80	18.15	16.50
	50	37.62	35.64	33.64	31.68	29.68	27.72	25.76	23.76	21.98	19.80
	40	47.02	44.55	42.07	39.60	37.12	34.65	32.17	29.70	27.22	24.75
	30	62.70	59.40	56.10	52.80	49.50	46.20	42.86	39.60	36.30	33.00
Initial Press. 150 Lbs. G. Vacuum 2 Lbs. Abs.	25	68.40	64.80	61.20	57.60	54.00	50.04	46.80	43.20	39.60	36.00
	20	85.50	81.00	76.50	72.00	67.50	63.00	58.50	54.00	49.50	45.00
	18	95.00	90.00	85.00	80.00	75.00	70.00	65.00	60.00	55.00	50.00
	16	106.87	101.25	95.62	90.00	84.37	78.75	73.12	67.50	61.87	56.25
	15	114.00	108.00	102.00	96.00	90.00	84.00	78.00	72.00	66.00	60.00
	14	122.14	115.71	109.29	102.86	96.43	90.00	83.57	77.14	70.71	64.28
	13	131.53	124.61	117.61	110.77	103.84	96.92	90.00	83.07	76.15	69.22
	12	142.50	135.00	127.50	120.00	112.50	105.00	97.50	90.00	82.50	75.00
	11	155.45	147.30	139.00	130.90	122.70	114.50	106.30	98.16	90.00	81.80
	10	171.00	162.00	153.00	144.00	135.00	126.00	117.00	108.00	99.00	90.00

At the intersection of vertical column "85" and horizontal column 100 of Table 68, we find 16.82 millions.

Tables 69 and 70 give the maximum theoretical height to which pumps may lift water by suction at different temperatures. In practice these figures cannot be realized. It is customary to have the water gravitate to the pump for all temperatures over 120 degrees F.

TABLE 69.

MAXIMUM HEIGHT TO WHICH PUMPS CAN RAISE WATER BY SUCTION.

(Temperature of Water 40 degrees F.; Barometer 29.92.)

Vacuum in Suction Pipe, Inches of Mercury.	Theoretical Lift.	Probable Actual Lift.	Vacuum in Suction Pipe, Inches of Mercury.	Theoretical Lift.	Probable Actual Lift.
	Feet.	Feet.		Feet.	Feet.
1	1.1	0.9	16	18.0	14.4
2	2.2	1.8	17	19.1	15.3
3	3.3	2.7	18	20.2	16.1
4	4.5	3.6	19	21.4	17.1
5	5.6	4.5	20	22.5	18.0
6	6.7	5.4	21	23.7	18.9
7	7.9	6.3	22	24.8	19.8
8	9.0	7.2	23	25.9	20.7
9	10.1	8.1	24	27.0	21.6
10	11.3	9.0	25	28.2	22.7
11	12.4	9.9	26	29.3	23.9
12	13.5	10.8	* 27	30.4	24.3
13	14.6	11.7	28	31.6	25.2
14	15.8	12.6	29	32.7	26.1
15	16.9	13.5	† 29.68	33.6	....

* Vacua greater than 27 inches are practically unobtainable in pumping practice except in connection with condensers.

† Maximum theoretical vacuum obtainable with water at 40 degrees F. and barometer of 29.92 inches.

TABLE 70.

MAXIMUM THEORETICAL HEIGHT TO WHICH A PUMP CAN LIFT WATER BY SUCTION AT DIFFERENT TEMPERATURES.

(Barometer 29.92.)

Temperature of Feed Water.	Maximum Theoretical Lift.	Temperature of Feed Water.	Maximum Theoretical Lift.
Degrees F.	Feet.	Degrees F.	Feet.
40	33.6	130	29.2
50	33.5	140	27.8
60	33.4	150	25.4
70	33.1	160	23.5
80	32.8	170	20.3
90	32.4	180	16.7
100	31.9	190	12.8
110	31.3	200	7.6
120	30.3	210	1.3

275. Size of Boiler-Feed Pump.

Let D = diameter of water cylinder, inches.
d = diameter of the steam cylinder, inches.
L = length of stroke, inches.
N = number of working strokes per minute.
H = head in feet between suction and boiler water level.
R = resistance in pounds per square inch between suction level and boiler water level due to valves, pipes, and fittings.
p = boiler pressure, pounds per square inch.
S = ratio of the water actually delivered to the piston displacement.
W = weight of water delivered, pounds per hour.
I = indicated horse power of the pump at maximum capacity.
E = mechanical efficiency of the pump, taken as the ratio of the water horse power at the discharge opening to the indicated horse power of the pump, steam end.

Then

$$W = \frac{\pi}{4} \cdot \frac{D^2}{144} \cdot \frac{LN}{12} \times 60 \times 62.5 \times S = 1.7\, D^2LNS. \quad (132)$$

$$D = 0.77 \sqrt{\frac{W}{LNS}}. \quad (133)$$

$$d = D\sqrt{\frac{p + R + 0.433\, H}{Ep}}. \quad (134)$$

$$I = \frac{W(p + R + 0.433\, H)\, 2.3}{33{,}000 \times 60 \times E}. \quad (135)$$

In average practice the piston or plunger displacement is made about twice the capacity found by calculation from the amount of water required for the engine, to allow for leakage, steam consumption of the auxiliaries, blowing off, and pump slip.

For pumps with strokes of 12 inches or over, the speed of the plunger or piston is usually limited to 100 feet per minute as a maximum to insure smooth running. For shorter strokes a lower limit should be used. The maximum number of strokes ranges from 100 for strokes over 12 inches in length, to 200 for strokes under 5 inches. Boiler-feed pumps should be designed to give the desired capacity at about one-half the maximum number of strokes or less.

Pump slip varies from 5 to 40 per cent, depending upon the condition of the piston and valves and the number of strokes. An average value for piston and plunger pumps in first-class condition is 8 per cent when

operating at rated capacity, but it is wise to allow a much larger figure, say 20 per cent, for leakage caused by wear.

The area of the steam cylinder is made from 2 to 2.5 times that of the water end to allow for the various friction losses and the drop in pressure between the pump throttle and the boiler. The total head pumped against includes the suction lift, the friction of valves and fittings, the distance between the suction inlet and the boiler level, and the boiler pressure. The excess head varies in practice from 15 to 40 per cent of the boiler pressure; an average figure is 25 per cent. In allowing for the drop in steam pressure between boiler and pump a liberal figure is 25 per cent.

The application of formulas (132) to (135), including the practical considerations stated above, is best illustrated by a specific example.

Example: Determine the size of direct-acting single-cylinder feed pump necessary to supply water to 1000 horse power of boilers. Gauge pressure 100 pounds per square inch; feed-water temperature 150 degrees F.

One horse power is equivalent to the evaporation of 34.5 pounds of water from and at 212 degrees F.; but the pump is usually designed to supply about twice the capacity.

Thus $W = 62{,}400$ (under the given conditions).
$S = 0.8$ (by assumption).
$LN = 1200$ (on the basis of 100 feet per minute).

Substitute these values in (133):

$$D = 0.77\sqrt{\frac{62{,}400}{1200 \times 0.8}} = 6.2 \text{ inches, — call it 6 inches,}$$

since the assumptions have been very liberal.

Assume $(.433\ H + R) = 0.25\,p$ and $E = 0.65$.

Substitute these values in (134):

$$d = 6\sqrt{\frac{100 + 25}{0.65 \times 100}}$$
$$= 8.35, \text{ — call it 8.5 inches.}$$

Allowing 100 strokes per minute the length of the stroke must be

$$L = 1200 \div 100 = 12 \text{ inches.}$$

The dimensions of the pump are $8\frac{1}{2} \times 6 \times 12$.

The indicated horse power at maximum load may be obtained by substituting the proper values in (135), thus:

$$I = \frac{62{,}400\ (100 + 25)\ 2.3}{33{,}000 \times 60 \times 0.65}$$
$$= 13.9 \text{ I.H.P.}$$

276. Steam-Pump Governors. — Fig. 276 shows a section through a Fisher pump governor, illustrating a device for maintaining a practically constant pressure in the discharge pipe irrespective of the quantity of water flowing. It embodies a pressure-reducing valve in the steam supply pipe of the pump, actuated by the slight variations in water pressure. When the demand for water increases, the pressure in the discharge pipe tends to decrease, and this drop in pressure (transmitted to the pump governor by suitable piping) causes more steam to be admitted, which increases the speed of the pump. The governor is connected to the steam inlet of the pump at B and the steam enters at A. Double-balanced valve C regulates the supply of steam to the cylinder by the amount it is raised from the seat. The valve is held open by spring G, the compression of which may be regulated by hand wheel K. The water pressure from the discharge pipe acts on piston F and tends to overcome the resistance of the spring. The difference in pressure between the water and the spring determines the position of valve C.

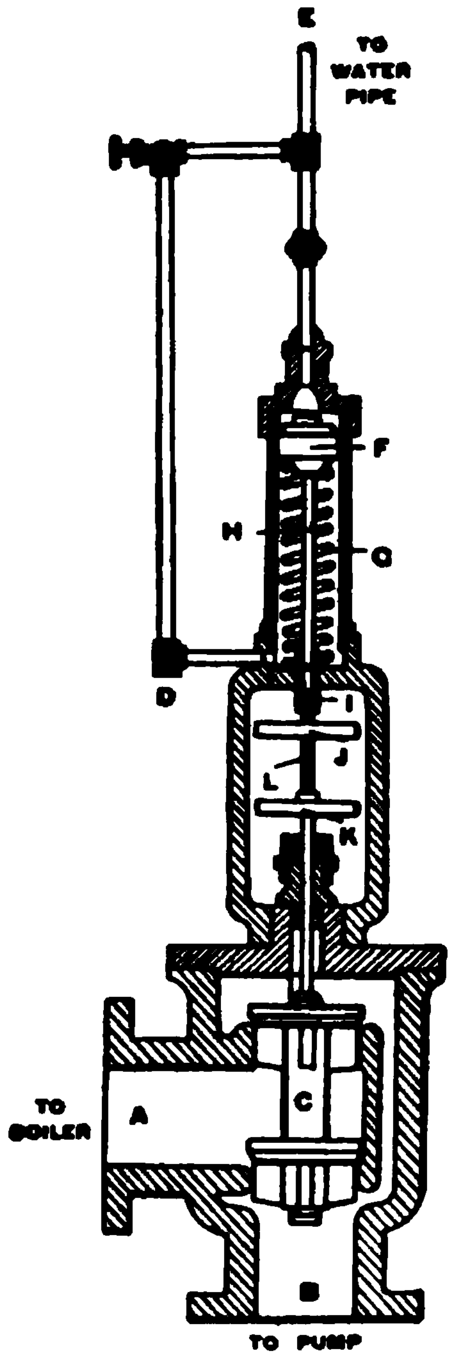


FIG. 276. Fisher Pump Governor.

Piston rod H is pinned to sleeve I and valve stem L screwed into this sleeve by means of hand wheel K. Hence during ordinary operation the piston, piston rod sleeve, valve stem, and valve act as a single unit. By turning the hand wheel K, valve stem L will screw into sleeve I and the tension on the spring will be increased. Hand wheel J serves as a lock nut and prevents K from turning during normal operation.

277. Feed-Water Regulators. — The water level in the boiler should be kept as nearly constant as possible, and this necessitates considerable attention on the part of the fireman, especially with fluctuating loads. There are a number of devices on the market which are designed to automatically maintain a constant level, and in many small plants where the duties of the fireman are numerous such devices in connection with high and low water alarms are of considerable assistance. Their action, however, is not always positive on account of wear or sticking of parts, and engineers as a rule prefer to rely upon hand regulation. In large stations regulators are seldom used.

Fig. 277 shows a section through a Kitts feed-water regulator, consisting of two parts, the chamber F and the regulating valve V. The

float chamber is connected to the boiler or water column at O and E, and the regulating valve to the feed main at R and to the boiler feed pipe at W. When the water in the boiler falls below the mean level, the weight B overcomes the counterweight G and closes needle valve L by means of compound levers. At the same time an extension on valve L lifts spring A and opens exhaust valve D. This removes the steam

FIG. 277. Kitts Feed-Water Regulator.

FIG. 278. Rowe Feed-Water Regulator.

pressure from the top of diaphragm C, in the regulating valve, through the agency of pipe K. The pressure from the pump raises the disk T and water flows into the boiler until the water rises to the mean level. When weight B becomes submerged its weight is overcome by counterweight G, valve L is opened and exhaust valve D is closed. This admits steam pressure to the diaphragm C and forces disk T to its seat, cutting off the supply of water to the boiler.

The Rowe feed-water regulator, Fig. 278, depends for its operation on a familiar float-controlled valve mechanism. The vessel A is connected to the boiler above and below the water line, and the float C, following the water level up and down, actuates a balanced valve in accordance with the boiler-feed requirements. When this apparatus is used to regulate the feed of a single boiler the opening G in the valve chamber is connected to the steam space of the boiler and the outlet H

is carried to the steam inlet of the feed-water pump. When the water level is normal the float closes the valve L and thereby cuts off the supply of steam to the pump cylinders. Communication between chambers A and R is prevented by means of a diaphragm M. When the water level falls below normal the float pulls the valve down, opening the way for steam to pass from the inlet G to the outlet H and thence to the pump. When the regulator is used to control a battery of boilers the pump discharge delivers into the inlet G and the water

FIG. 279. A Typical Geared Triplex Pump.

passes through H to the boiler-feed main. Should the water level fall beyond a predetermined limit by reason of any accidental discontinuance of the water supply which the apparatus cannot correct, the float would open the valve F of the alarm whistle O mounted on the top of the main vessel.

278. Power Pumps. — Piston pumps, geared, belted, or direct connected to electric motors, gas engines, and water motors, are used chiefly where steam power is not available. Their general utility is evidenced by the rapidly increasing number installed in situations formerly occupied by the direct-acting steam pump. The efficiency of

this type of pump depends in a large measure upon the character of the driving motor and the efficiency of the transmitting mechanism. High-speed power pumps direct connected to electric motors give

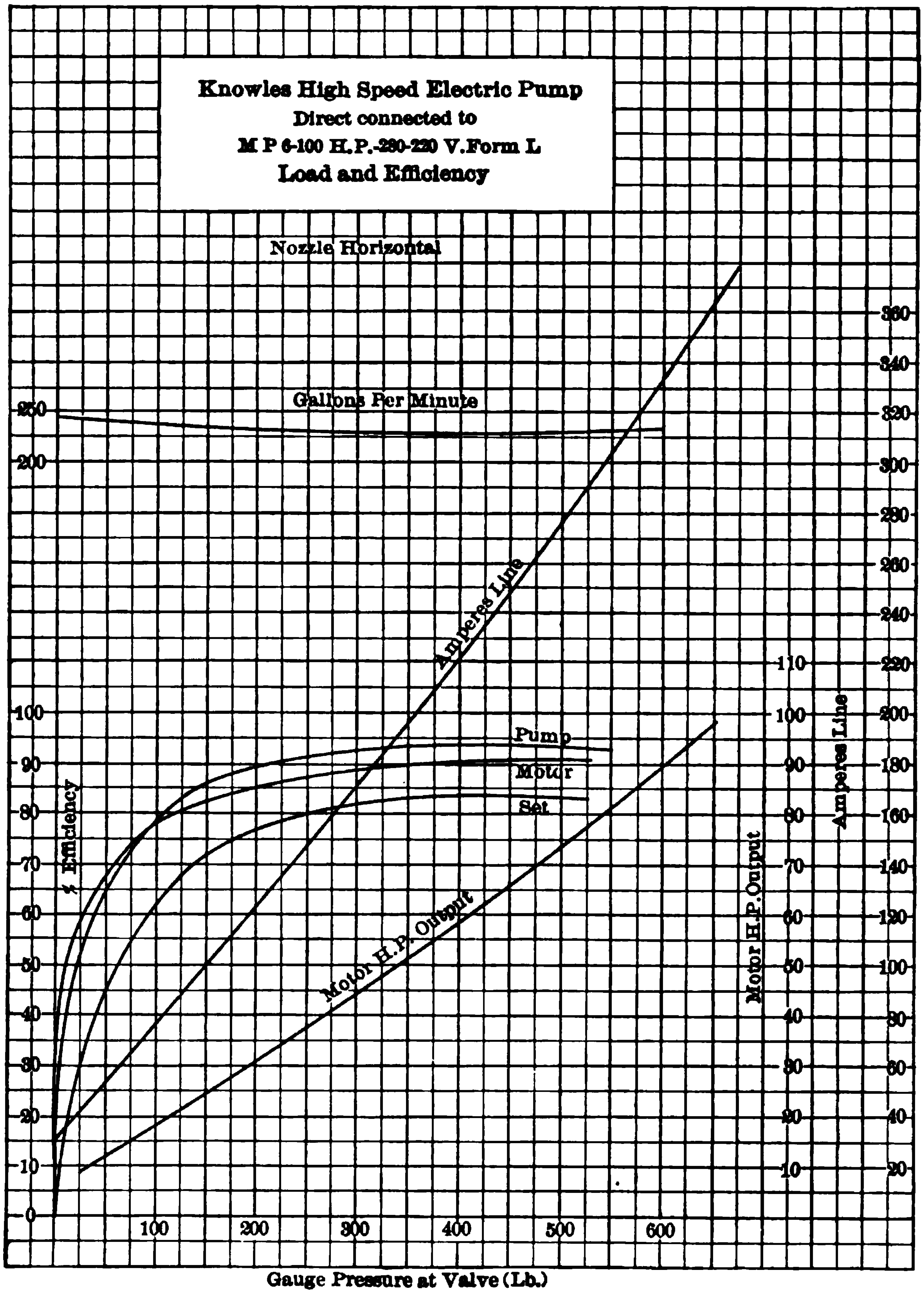
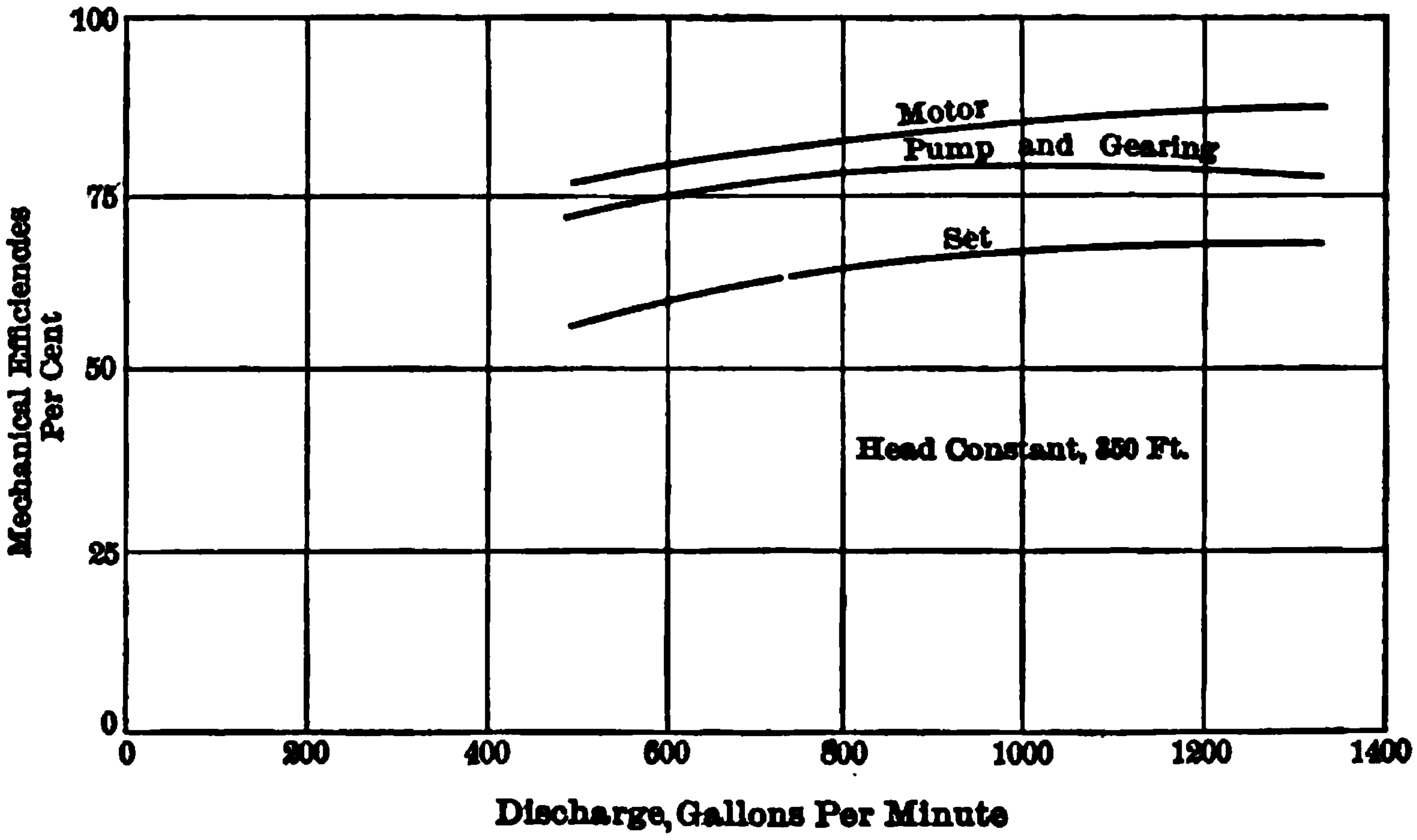


Fig. 279a.

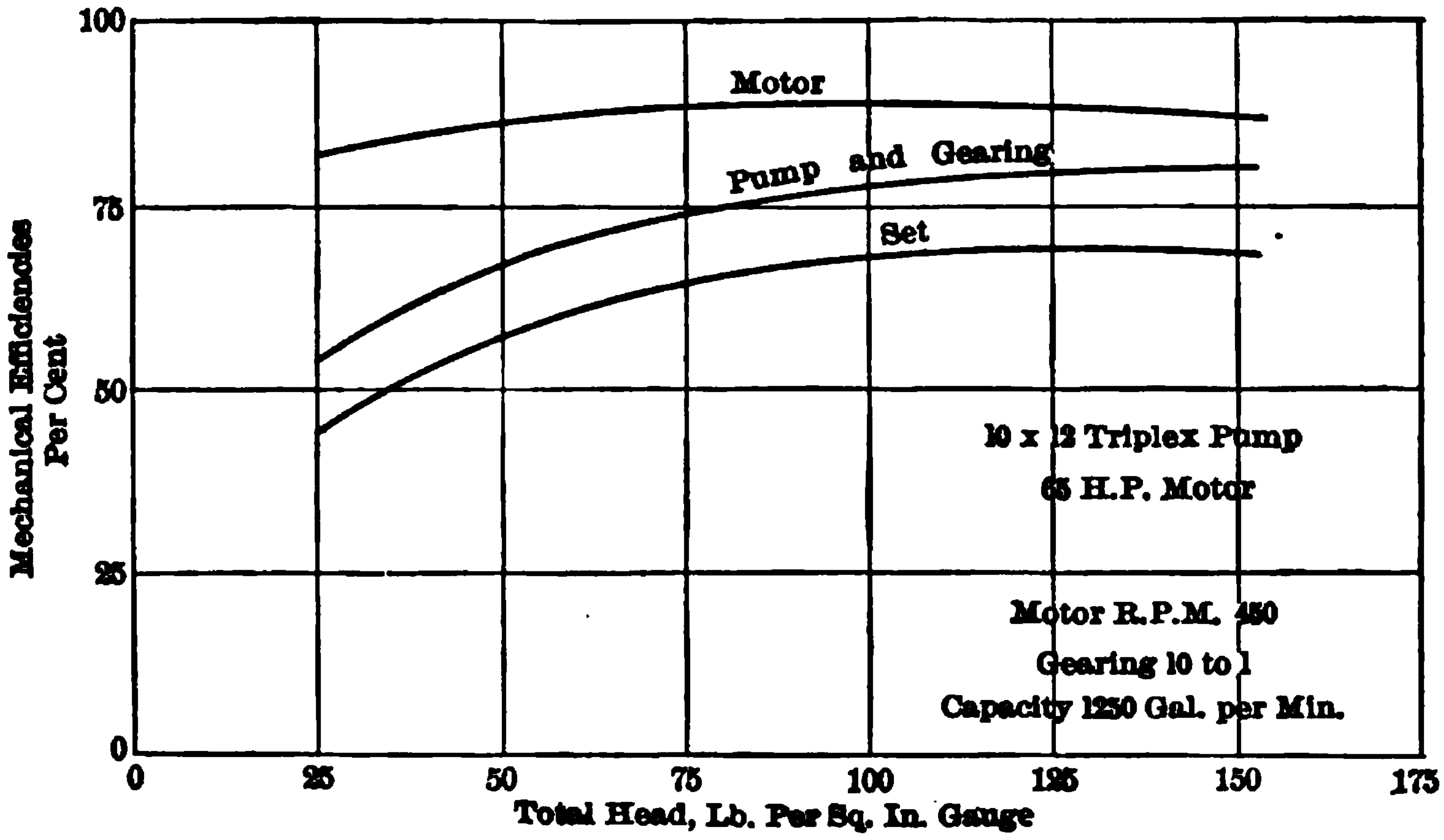
efficiencies from line to water horse power as high as 83 per cent, while the low-speed geared type seldom exceed 70 per cent. The curves in Fig. 279a give the performance of a direct-connected triplex pump,

and those in Fig. 280 the performance of a triplex pump geared to an electric motor. Both of these performances are exceptionally good and are considerably above the average.

Power Pumps: Eng. Mag., Jan., 1905, p. 616; Col. Guard, Nov. 17, 1905; Elec. Rev., Jan. 10, 1902; Elec., N.Y., Oct. 12, 1904; Elec. World, Oct. 14, 1905, p. 667; Engr., Lond., Oct. 17, 1902, p. 377; Engineering, Sept. 1, 1905, p. 275; Engr. U.S., Jan. 1, 1904, p. 47.



Head Constant. Speed Variable.



Speed Constant. Head Variable.

FIG. 280. Performance of a 65-Horse-Power, Motor-Driven Triplex Pump. Geared Type.

279. Injectors. — As a boiler feeder the injector is an efficient and convenient device, cheap and compact, with no moving parts, delivers hot water to the boiler without preheating, and has no exhaust

steam to be disposed of. Its adoption in locomotives is practically universal, but in stationary practice it is limited to small boilers or single boilers or as a reserve feeder in connection with pumps. The objections to an injector are its inability to handle hot water, the difficulty of maintaining a continuous flow under extreme variation of load, and the uncertainty of operation under certain conditions. Fig. 281 illustrates the simplest form of single-tube injector. Boiler steam

Fig. 281. Elementary Steam Injector.

is admitted at A and, flowing through nozzle and combining tube to the atmosphere through G, partially exhausts the air from pipe B, thereby causing the water to rise until it comes in contact with the steam.

Fig. 282. Hancock Double-Tube Injector.

Fig. 283. Penberthy Automatic Injector.

The steam emerging from nozzle C at high velocity condenses on meeting the water and imparts considerable momentum to it. The energy in

the rapidly moving mass is sufficient to carry it across opening *O*, lift check *H* from its seat and force it into the boiler. The steam then ceases to escape at *G*.

280. Positive Injectors. — Fig. 282 shows a section through a Hancock injector, illustrating the principles of the double-tube positive type. Its operation is as follows: Overflow valves *D* and *F* are opened and steam is admitted, which at first passes freely through the overflow to the atmosphere and in so doing exhausts the air from the suction pipe. This causes the feed water to rise until it meets the jet of steam and the two are forced through the overflow. As soon as water appears at the overflow, valve *D* is closed, valve *C* partially opened, and valve *F* closed. This admits steam through the forcing jet *W* and, the overflow valves being closed, the water is fed into the boiler. In case the action is interrupted for any reason it is necessary to restart it by hand.

The chief advantage of the double-tube positive type lies in its ability to lift water to a greater height and to handle hotter water than the single-tube. Its range in pressure is also greater, that is, it will start with a lower steam pressure and discharge against a higher back pressure. Double-tube injectors are used almost exclusively in locomotive work.

281. Automatic Injectors. — Fig. 283 shows a section through the Penberthy injector. Its operation is as follows: Steam enters at the top connection and blows through suction tube *c* into the combining tube *d* and into chamber *g*, from which it passes through overflow valve *n* to the overflow *m*. When water is drawn in from the suction intake and begins to discharge at the overflow, the resulting condensation of the steam creates a partial vacuum above the movable ring *h* and the latter is forced against the end of tube *c*, cutting off the direct flow of water to the overflow. The water then passes into the boiler. Spill holes *i*, *i*, *i* are for the purpose of relieving the excess of water until communication with the boiler has been established. The action of opening and closing the overflow is entirely automatic. Where the conditions are not too extreme the automatic injector is to be preferred for stationary work because of its restarting features. It is also used on traction, logging, and road engines, where its certainty of action and special adaptability render it invaluable for the rough work to which such machines are subjected.

Injectors, Theory of: Trans. A.S.M.E., 10–339; Sibley Jour., Dec., 1897, p. 101; Power, May, 1901, p. 23; Thermodynamics of the Steam Engine, Peabody, Chap. IX; Theory of the Steam Injector, Kneass.

Injectors, General Description: Engr. U.S., Oct. 1, 1907, Nov. 15, 1907, July 15, 1904, p. 501, Feb. 2, 1903, p. 151; Power, Aug., 1906, p. 478; Engr., Lond., March 10, 1905, p. 244; Engineering, Aug. 30, 1895, p. 281.

282. Performance of Injectors. — The performance of an injector may be very closely determined from the equation

$$w = \frac{xr + q - t + 32}{t - t_0} \quad \text{(Kneass, "Theory of the Injector," p. 83).} \qquad (136)$$

in which

w = pounds of water delivered per pound of steam supplied.
x = quality of the steam supplied.
r = heat of vaporization.
q = heat of the liquid.
t = temperature of the discharge water.
t_0 = temperature of the suction water.

Figs. 284a, 284b, and 284c give the performance of a Desmond automatic injector as tested at the Armour Institute of Technology. The results check very closely with those calculated from above equation. Referring to Fig. 284a it will be seen that the weight of water delivered per pound of steam decreases as the initial pressure is increased, all other factors remaining the same. From Fig. 284b it will be noted that the weight of water delivered per pound of steam decreases as the temperature of suction supply is increased up to a point where the injector " breaks " or becomes inoperative. This critical temperature varies with the different types of injectors, being highest for the double-tube type, but seldom exceeds 160 degrees F. Fig. 284c shows that the weight of water delivered per pound of steam is practically constant for all discharge pressures within the limits of the apparatus.

Table 71 gives the range of working steam pressures for standard " Metropolitan " injectors with varying suction heads and temperatures, and, though strictly applicable to this particular type only, is characteristic of all makes.

In selecting an injector the following information is desirable for best results:

1. The lowest and highest steam pressure carried.

2. The temperature of the water supply.

3. The source of water supply, whether the injector is used as a lifter or non-lifter.

4. The general service, such as character of the water used, whether the injector is subject to severe jars, etc.

Injectors, Tests of: Eng. News, March 17, 1898, July 16, 1896, p. 39; Locomotive Engineering, May, 1900, p. 204; Power, Oct., 1904, p. 602; Railroad Gazette, Dec. 11, 1896; Thermodynamics of the Steam Engine, Peabody, Chapter IX; Theory of the Injector, Kneass.

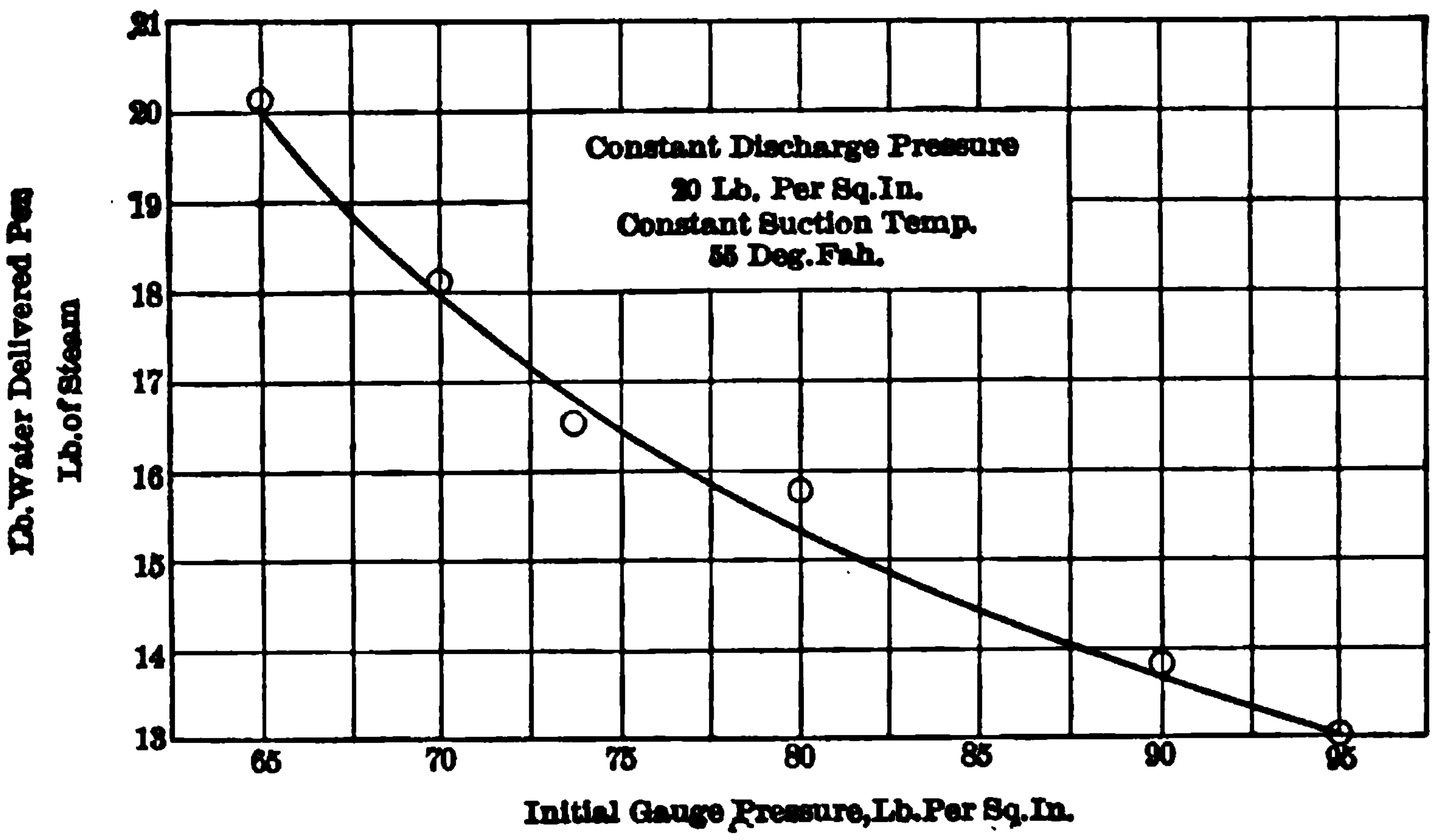


FIG. 284a. Performance of an Automatic Injector with Varying Initial Pressure.

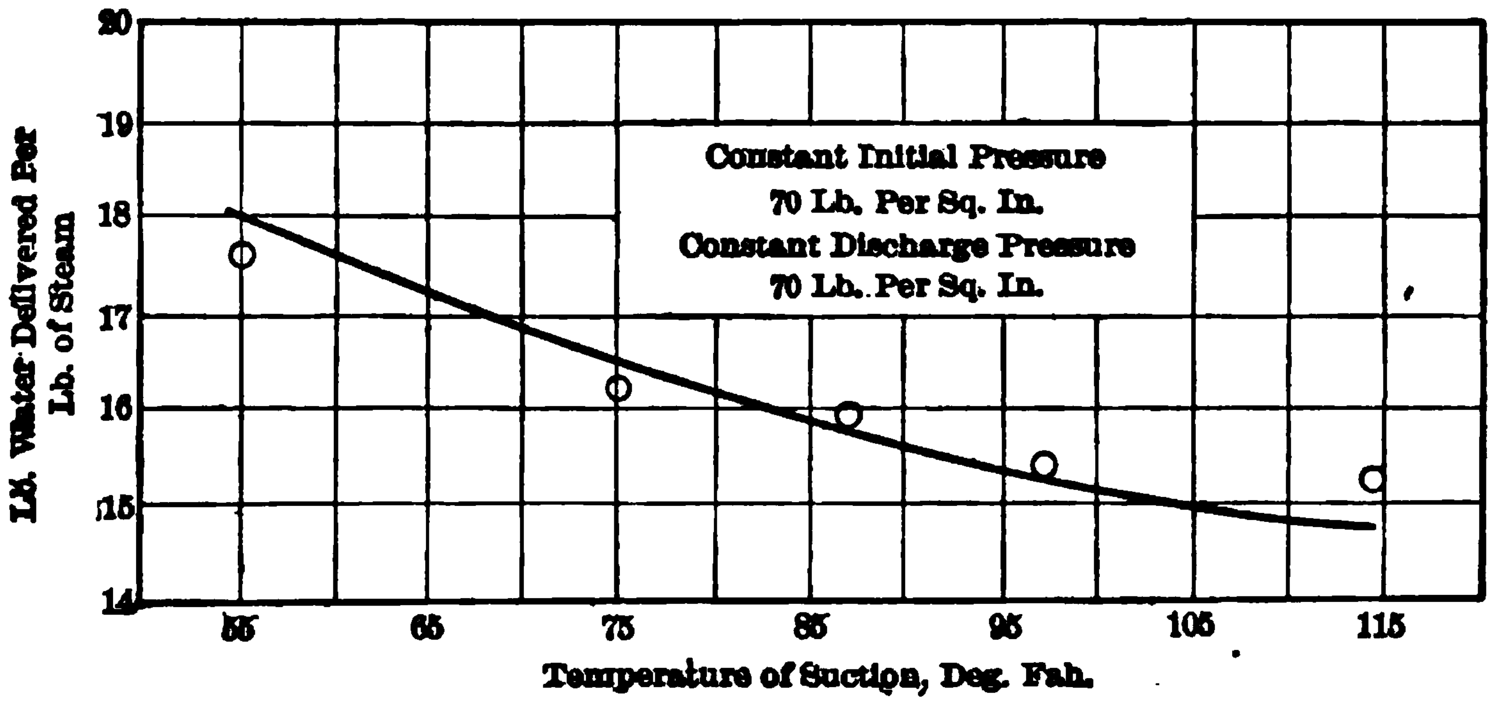


FIG. 284b. Performance of an Automatic Injector with Varying Suction Temperature

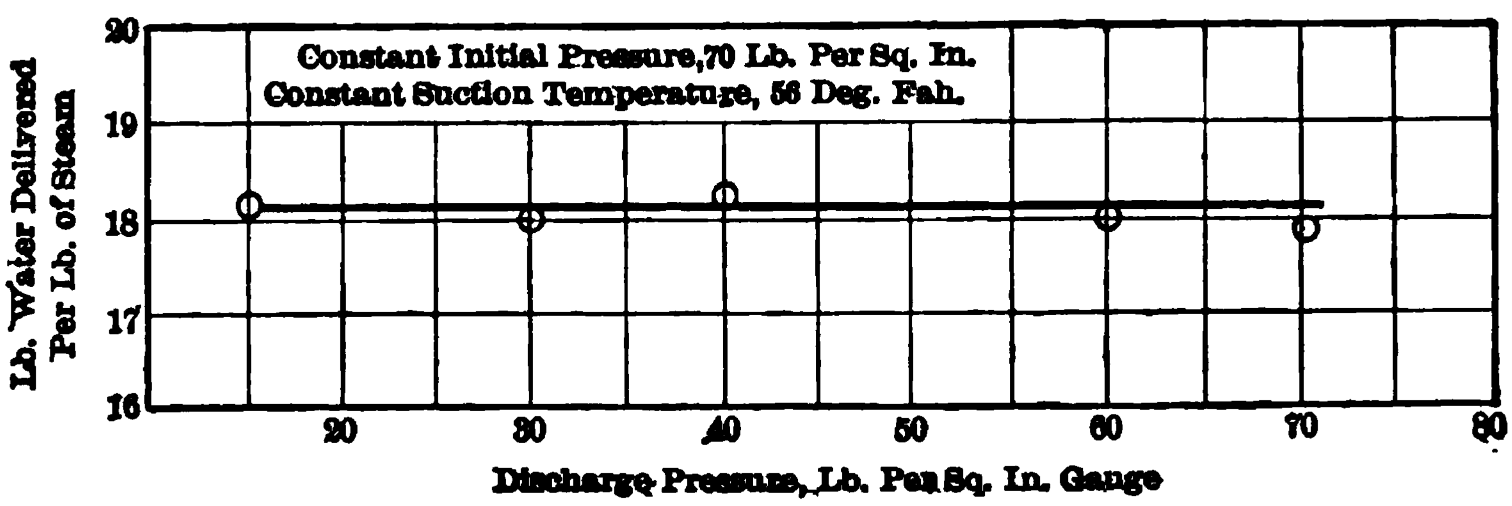


FIG. 284c. Performance of an Automatic Injector with Varying Discharge Pressure.

TABLE 71.

RANGE IN WORKING PRESSURES.

Standard " Metropolitan " Steam Injectors.

Suction Temperature, Degrees F.	Automatic.				
	Suction Head, Feet.				
	2	8	14	20	Under Pressure.
Under 60	25 to 150	30 to 130	42 to 110	55 to 85	20 to 160
100	26 to 120	33 to 100	55 to 80	....	25 to 125
120	....	....	....	....	26 to 85
140	....	....	....	....	....

Suction Temperature, Degrees F.	Double Tube.				
	Suction Head, Feet.				
	2	8	14	20	Under Pressure.
Under 60	14 to 250	23 to 220	27 to 175	42 to 135	14 to 250
100	15 to 210	26 to 160	37 to 120	46 to 70	15 to 210
120	20 to 185	30 to 120	42 to 75	....	20 to 185
140	20 to 120	35 to 70	....	....	20 to 120

283. Injector *vs.* Steam Pump as a Boiler Feeder.— From a purely thermodynamic standpoint the efficiency of an injector is nearly perfect, since the heat drawn from the boiler is returned to the boiler again, less a slight radiation loss. As a pump, however, the injector is very inefficient and requires more fuel for its operation than very wasteful feed pumps. This is best illustrated by an example: An injector of modern construction will deliver say 15 pounds of water to the boiler per pound of steam supplied, with delivery temperature of 150 degrees F. This corresponds to a heat consumption of 71 B.T.U. per pound of water delivered, thus:

With initial pressure of 115 pounds absolute,

$$\lambda = 1185.$$

Heat in the water delivered to the boiler,

$$150 - 32 = 118 \text{ B.T.U. above 32 degrees F.}$$

Heat of 1 pound of steam above a feed temperature of 150 degrees F.

$$1185 - 118 = 1067 \text{ B.T.U.}$$

Heat required to deliver 1 pound of water to the boiler,

$$\frac{1067}{15} = 71 \text{ B.T.U.}$$

A simple direct-acting duplex pump consumes say 200 pounds steam per I.H.P. hour. Assume the extreme case where the exhaust steam will not be used for heating the feed water and the latter is fed into the boiler at 60 degrees F.

The heat supplied to the pump per I.H.P. hour,

$$200 \{1185 - (60 - 32)\} = 231{,}400 \text{ B.T.U.}$$

Assuming the low mechanical efficiency of 50 per cent, the heat required to develop one horse power at the water end will be

$$231{,}400 \div 0.50 = 462{,}800 \text{ B.T.U. per hour.}$$

Since the steam pressure is 100 pounds gauge, the equivalent head of water at 60 degrees F. is

$$2.3 \times 100 = 230 \text{ feet.}$$

Assume the friction in the feed pipe, the resistance of valves, etc., to be 30 per cent of the boiler pressure; the total head pumped against will be

$$230 + 69 = 299, \text{ say } 300 \text{ feet.}$$

$$1 \text{ horse-power hour} = 1{,}980{,}000 \text{ foot-pounds per hour.}$$

$$\frac{1{,}980{,}000}{300} = 6600 \text{ pounds,}$$

that is, 1 horse power at the pump will deliver 6600 pounds of water per hour to the boiler against a head of 300 feet.

The heat consumption per pound of water delivered,

$$\frac{462{,}800}{6600} = 70.1 \text{ B.T.U.}$$

If the feed water is heated to say 210 degrees F. by the exhaust steam from the pump, the heat consumption will be 55.5 B.T.U. as against 70.1 without the heater.

Thus even in this extreme case of poor steam-pump performance the heat consumption lies in favor of the pump. With the better

grades of pumps this disparity is considerably greater, and decidedly so if the exhaust steam is used to preheat the feed water. For intermittent operation the condensation losses in the pump may more than offset this gain. Other conditions, however, such as compactness, low first cost, and ease of operation are oftentimes considerations, and the heat consumption is of minor importance.

284. Air Pumps. — Condenser air pumps may be divided into two classes:

1. Wet-air pumps and
2. Dry-air pumps.

The former handle both air and water and the latter air alone. Ordinary jet-condenser wet-air pumps handle simultaneously the circulating water, condensed steam, and entrained air, and are, in fact, a combination of circulating and vacuum pump. Surface-condenser wet-air pumps are the same in principle and design, but are smaller in size for a given main engine output, as they handle the condensed steam and air only.

Wet-air pumps may be driven by the main engine or independently and may be direct acting, Fig. 203, or fly wheel driven, Fig. 228. The fly-wheel type may be steam, electric, or belt driven. Dry-air pumps are virtually air compressors, as their function is to compress air from the pressure existing in the condenser to that of the atmosphere. They are generally of the fly-wheel type. Where a high degree of vacuum is necessary the air cylinders are compounded, as efficient compression of say ½ pound to 15 pounds or 30 to 1 is too great for a single stage.

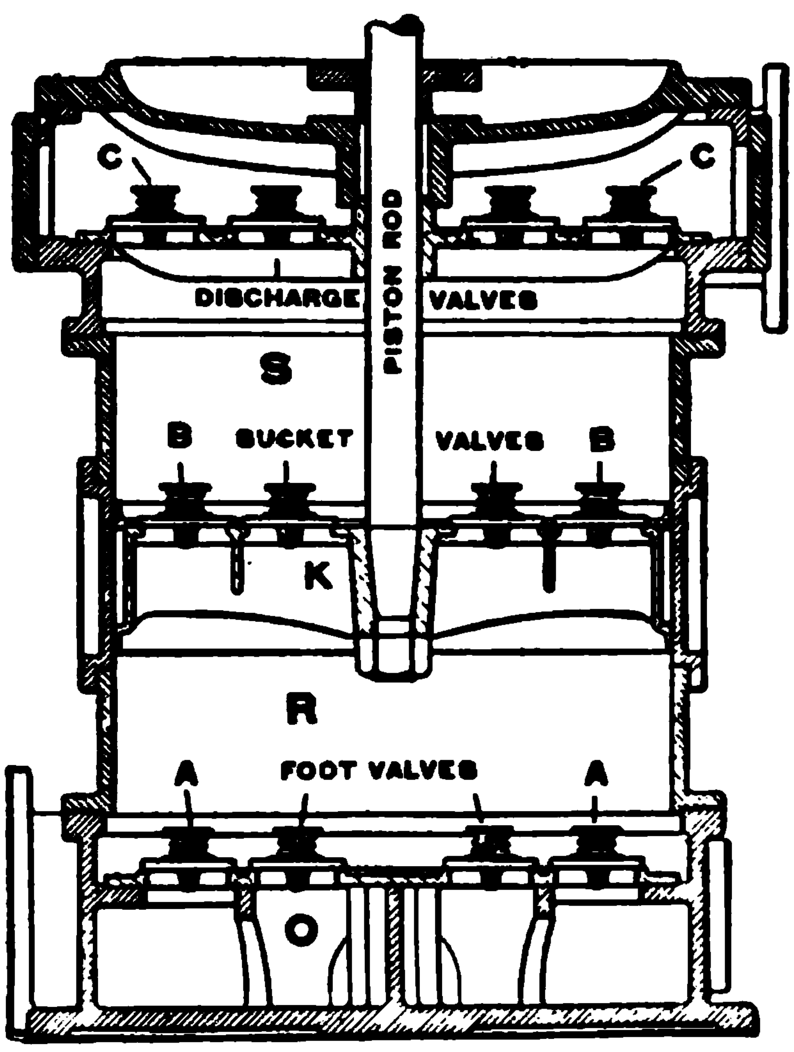


FIG. 285. Dean Air Pump.

285. Dean Air Pump. — Fig. 285 shows a section of the air cylinder of a Dean twin-cylinder wet-air pump as applied to a jet condenser. There are three sets of valves, the suction or foot valves *A*, *A*, the lifting or bucket valves *B*, *B*, and the head or discharge valves *C*, *C*. On the upward stroke of the piston or bucket a partial vacuum is formed in the chamber between the bucket and the lower head, causing the water

and air in the bottom of the barrel to lift the foot valves A, A from their seats and flow into the cylinder. On the downward stroke the foot valves A, A close and water and air are entrapped in chamber R between the lower head and the bucket. As the bucket descends, the pressure of air in the cylinder lifts the bucket valves B, B from their seats and permits the air and water to escape to the upper portion S of the cylinder between the head plate and the bucket. On the next upward stroke the water and air are forced through the discharge valves C, C into the hot well. This discharge of water and air from the top compartment is simultaneous with influx of water and air in the lower chamber.

See paragraph 209 for other types of wet-air pumps in connection with jet condensers.

286. Size of Wet-Air Pumps; Jet Condensers. — In proportioning such pumps the quantity of cooling water and condensed steam to be taken care of is readily determined, but the percentage of air mingled with it must be estimated. Surface water under atmospheric pressure ordinarily contains from 2 to 5 per cent of air by volume. To provide for possible leakage a very liberal factor is usually allowed, an average figure being about 10 per cent.

Let Q = total volume of air and water in cubic feet per hour to be handled by the pump.
V = volume of cooling water in cubic feet per hour.
v = volume of condensed steam in cubic feet per hour.
v_a = volume of air at pressure P_a and temperature T_a.
$V + v$ = total volume of water and condensed steam at atmospheric temperature.
T_a = temperature of the air entering the condenser, degrees F.
T_2 = temperature of the discharge water, degrees F.
T_0 = initial temperature of the cooling water, degrees F.
P_a = atmospheric pressure, pounds per square inch.
P_c = total pressure in the condenser, pounds per square inch.
P_v = pressure of aqueous vapor at temperature T_2.

Then $(V + v)$ = volume of water to be pumped from the condenser per hour and v_a = volume of air at atmospheric pressure and temperature T_a entering the condenser; but on entering the condenser the air is increased in volume, due to the reduction in pressure and the increase in temperature, and the total volume to be exhausted per hour by the pump is

$$Q = V + v + v_a \frac{P_a}{P_c - P_v} \times \frac{T_a + 460}{T_0 + 460}. \qquad (138)$$

Under average conditions of reciprocating engine practice $v = \frac{1}{25} V$, $P_a = 15$ pounds per square inch, $P_c = 2$ pounds per square inch, $P_v = 1.27$, $T_2 = 110$ degrees F., $T_0 = 60$ degrees F. Substituting these values in above equations, assuming $v_a = 10\%$ of V, we get $Q = 3.4\ V$.

Average practice gives 3 V as the pump displacement per hour for a single-acting pump and 3.5 V for a double-acting pump, the cylinders being ordinarily proportioned on a basis of 50 feet per minute piston velocity at rated capacity.

Table 72 gives the approximate sizes of air pumps for condensers as manufactured by prominent makers.

TABLE 72.

APPROXIMATE SIZES OF AIR PUMPS FOR CONDENSERS.

Pounds of Steam Condensed per Hour.	Jet Condenser.			Surface Condensers.	
	Duplex Pump.	Horizontal Double Acting Pump.	Vertical 2-Cylinder Single Acting.	Horizontal.	Vertical 2-Cylinder.
500 to 1,000	$4\frac{3}{4} \times 5$	6×7	5×4	$3\frac{1}{2} \times 4$	....
1,000 to 1,500	$5\frac{3}{4} \times 6$	8×7	6×4	4×4	....
1,500 to 2,000	$6\frac{1}{2} \times 6$	8×12	7×5	4×6	....
2,000 to 2,500	$7\frac{1}{2} \times 6$	9×9	9×6	5×7	....
2,500 to 3,000	7×10	9×10	10×8	5×8	....
3,500 to 4,000	8×10	11×12	11×9	6×8	....
4,000 to 4,500	$8\frac{1}{2} \times 10$	12×14	12×8	7×9	....
4,500 to 5,000	9×10	14×14	12×10	7×10	....
5,000 to 6,000	10×10	14×16	14×10	7×12	....
6,000 to 7,000	$10\frac{1}{2} \times 10$	15×16	15×10	8×10	8×4
7,000 to 8,000	11×10	15×18	15×12	8×12	8×6
8,000 to 9,000	12×10	16×18	16×10	9×12	9×6
9,000 to 10,000	12×15	18×18	17×12	10×12	10×8
10,000 to 15,000	15×15	20×24	20×12	12×14	11×8
15,000 to 20,000	17×15	24×24	22×15	14×16	12×8
20,000 to 25,000	19×15	26×24	24×18	16×24	14×10

Wet-air pumps are usually independently driven, making it possible to vary the speed of the pump irrespective of the engine speed and to create a vacuum before starting the engine. Occasionally, however, when the load is constant, as in pumping-engine practice, the pump may be driven by the main engine.

Air Pumps: Power, Nov., 1904, p. 652; Engineering, Nov. 17, 1905; Engr. U.S., Jan. 1, 1906, Elec. Engr., Lond., Nov. 14, 1902; American Mach., Jan. 31, 1901, p. 113; National Engr., June, 1906.

Air Pumps. — Reference Books: Hausbrand, Evaporating and Condensing Apparatus, p. 383; Whitham, The Steam Engine, p. 301; Thurston, Manual of the Steam Engine, Vol. II, p. 145; Seaton, Manual of the Marine Engine, p. 301.

287. Edwards Air Pump. — Fig. 286 shows a section through the air cylinder of an Edwards air pump. This device belongs to the surface-condenser "wet-air pump" class, as both the water of condensation and the entrained air are exhausted simultaneously by the same piston. Unlike the standard type of wet-air pumps, foot valves and bucket valves are entirely dispensed with. The condensed steam flows continuously by gravity from the condenser into the base of the pump through passage *A* and annular space *B*. As the piston *C* descends it forces the water from the lower part of the casing *F* into the cylinder proper through the ports *P*, *P*. On the upward stroke the ports in the piston are closed and the air and water discharged through head valves *D* and exhaust port *E* to the hot well. The seats of valves *D* are constructed with a rib between each valve and a lip around the outer edge, so that each valve is water-sealed independently of the others. In earlier air pumps of this general type the clearance between the bucket and head valve seat is necessarily large, due to the space occupied by the bucket valves and the ribs on the under side of the valve seating. This clearance space reduces the capacity of the pump, since the air above the bucket must be compressed above atmospheric pressure before it can be discharged, and on the return stroke will expand and occupy a space which should be available for a fresh supply of air from the condenser. In the Edwards air pump the clearance space is reduced to a minimum, since there are no bucket valves to limit it. The absence of suction or foot valves still further increases the capacity of the pump for similar reasons. These pumps are arranged either single, double, or triplex; steam, electric, or belt driven; slow or high speed. They are ordinarily used in connection with surface condensers.

FIG. 286. Edwards Air Pump.

Edwards Air Pump: Engr., July 1, 1903, p. 536; Engineering, 62–221, 63–60, 64–767, 80–328; Eng. News, June 12, 1902, p. 478.

Centrifugal Wet Vacuum Pump: Power & Engr., Jan. 4, 1910.

288. Mullan Valveless Air Pump. — Fig. 287 gives several views of the "Mullan valveless air pump" as used in connection with the

Fig. 287. Mullan Valveless Dry-Air Pump.

C. H. Wheeler Company's "high-vacuum" condensing outfit. The pump is double acting and devoid of suction valves. The cylinder

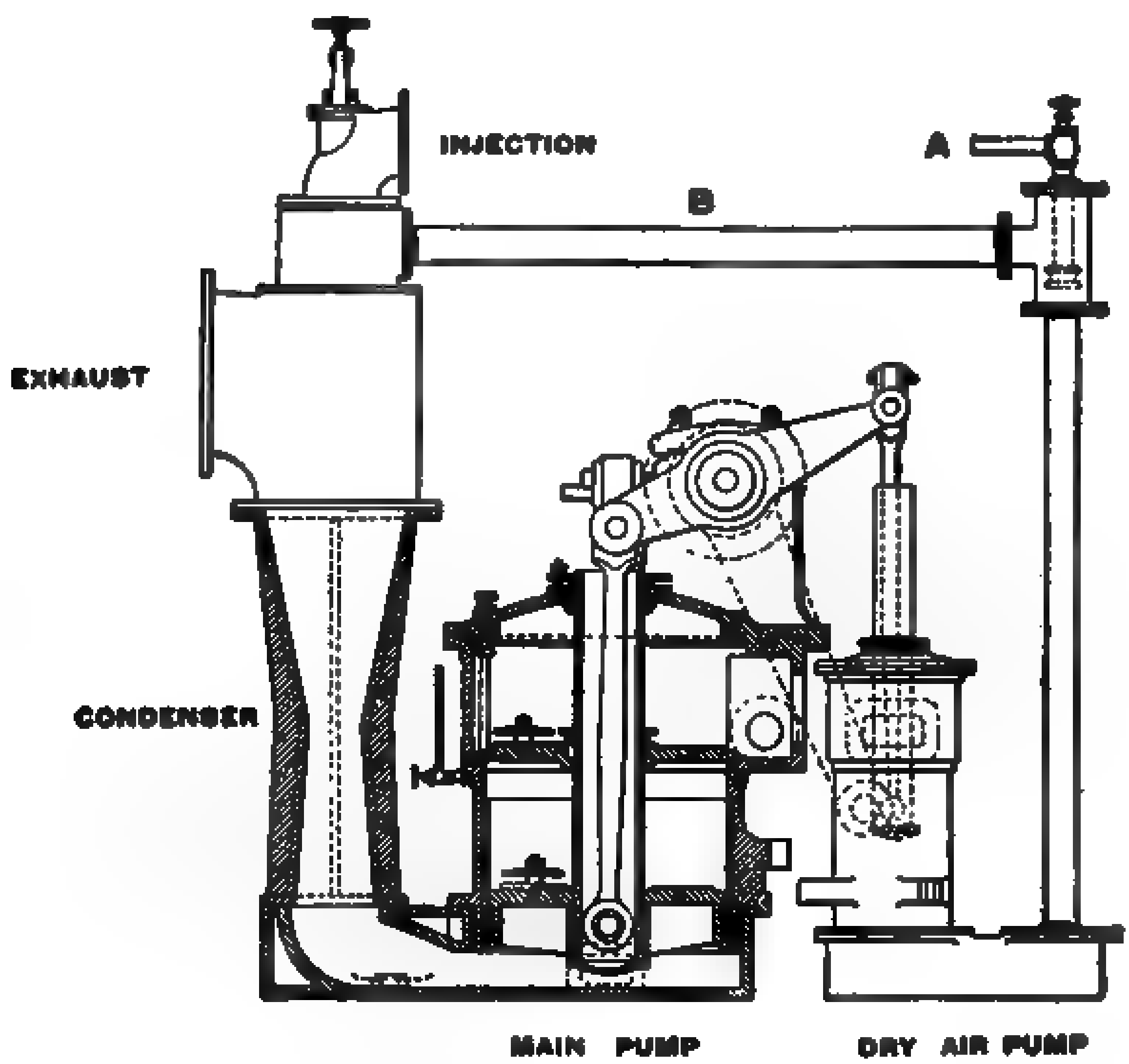


Fig. 288. Hewes and Phillips Air Pumps.

has a central port which is uncovered by the piston at each end of the stroke and covered at all other positions. Discharge valves of

the familiar spring-seated poppet type are located in both heads of the cylinder. As the piston moves from one end of its stroke to the other it forms a vacuum behind it and forces out the gases and water ahead of it; when it reaches the end of the stroke the central inlet port is uncovered and the vacuum behind the piston draws in the condensation and gases from the condenser. This operation is repeated on the return stroke.

The makers claim that the pump will operate, under shop test conditions, within one-half inch of the barometer, enabling them to guarantee a vacuum within two inches of absolute under full-load conditions of steam turbine operation.

289. Alberger Rotative Dry-Air Pump. — Fig. 289 shows a section through the air cylinder of an Alberger rotative dry-air pump, illus-

FIG. 289. Alberger Rotative Dry-Air Pump.

trating a type of pump in which the admission valve is mechanically operated. This pump is designed to operate with dry air only, all condensation being removed before the air enters the cylinder. This permits of the use of a small clearance space and makes it possible to run at a higher speed of rotation than can be secured with a type of pump in which water is used to seal the valves. Referring to Fig. 289, air is being taken into the right-hand end of the cylinder through inlet *A* and forced from the left-hand end through exhaust opening *B*. Rotary valve *O* mechanically opens to admission and mechanically closes the discharge. The discharge opening depends on the spring-regulated valve *C* at the top of the cylinder. Heads are water jacketed. Ports and passages are made large to reduce the friction of the air entering the pump, and to obviate the bad effects of clearance, an equalizing passage

is provided in valve O. The action of the passage is shown in the section to the right. When the piston reaches the end of the stroke the clearance space is filled with air at atmospheric pressure. If this pressure were not relieved the piston would travel a considerable distance before drawing in air from the condenser. By means of the equalizing passage the clearance space is connected to the opposite end of the cylinder and the vacuum there reduces the pressure in the clearance space.

290. Size of Wet-Air Pump for Surface Condenser. — Since the wet-air pump handles both the air and condensed steam, its theoretical capacity, neglecting clearance, may be determined by eliminating V from equation (138), which then becomes

$$Q = v + v_a \frac{P_a}{P_c - P_v} \times \frac{T_a + 460}{T_0 + 460}. \tag{139}$$

For the average reciprocating engine, $P_c = 2$, $T_2 = T_a = 110$, $T_0 = 60$, $P_v = 1.27$. Assuming $P_a = 15$ pounds per square inch, and substituting these values in (139), $Q = v + 25\, v_a$.

The volume of air entering the condenser varies so much with the character of the power plant equipment and the conditions of operation that any assumed "average" value of v_a may lead to serious error. (See Power & Engr., Feb. 2, 1909, p. 234.)

A study of some two hundred condenser installations gives

$Q = 10\, v$ for the average reciprocating engine and
$Q = 20\, v$ for average steam turbine practice.

Table 72 gives the cylinder dimensions of wet-air pumps as advocated by prominent condenser builders.

291. Size of Dry-Air Pumps. — "Dry-air" pumps are used in connection with barometric and surface condensers where a high degree of vacuum is essential, as in steam turbine practice. Such pumps are intended to exhaust the saturated non-condensable vapors only.

The capacity of the dry-air pump is based upon experience rather than theory. An investigation of some fifty installations gives

$Q = 20\, v$ to $30\, v$ for vacua under 27 inches.
$Q = 50\, v$ for vacua of 28 inches and over, both referred to a 30-inch barometer.

Professor Weighton states that "with suitable condenser arrangements and a reasonably air-tight system there is nothing gained in efficiency by the use of air pumps exceeding in capacity 0.7 of a cubic foot per pound of steam condensed up to a vacuum of 29 inches." (*Engineering Record*, May 19, 1906, p. 61.)

The work done by the average "high-vacuum" dry-air pump is a maximum for vacua between 18 and 20 inches.

This may be proved from Fig. 290, which represents a theoretical indicator card from the air-pump cylinder.

Let p_2 = pressure in the condenser, pounds per square inch absolute.
p_1 = atmospheric pressure.
v_2 = piston displacement, including clearance, cubic feet.
v_1 = volume of air in the cylinder when the valve opens to atmosphere, cubic feet.
v_c = clearance volume, cubic feet.

The work done is proportional to the area $ABCD$.

Area $ABCD$ = work done = area $EBIO+BAGI-FAGO-ECDF$.

Neglecting the exponential factor n for the sake of simplicity, thus making $pv = p_1v_1 = p_2v_2$ = constant,

$$W = \text{work done} = p_1v_1 + p_1v_1\int_{v_1}^{v_2}\frac{dv}{v} - p_2v_2 - p_1v_c + p_2v_c. \tag{140}$$

Substitute p_1v_1 for its equivalent p_2v_2 and $\frac{p_1v_1v_c}{v_2}$ for its equivalent p_2v_c and integrate.

$$W = p_1v_1 + p_1v_1 \log_e v_2 - p_1v_1 \log_e v_1 - p_1v_1 + \frac{p_1v_1v_c}{v_2}, \tag{141}$$

making the first derivative zero.

$$\frac{dw}{dv} = 0 = \log_e v_2 - 1 - \log_e v_1 + \frac{v_c}{v_2}, \tag{142}$$

$$0 = \log_e \frac{v_2}{v_1} - 1 + \frac{v_c}{v_2}, \tag{143}$$

i.e., W is a maximum when

$$-\log_e \frac{v_2}{v_1} = \frac{v_c}{v_2} - 1 \tag{144}$$

or

$$\log_e \frac{p_1}{p_2} = 1 - \frac{v_c}{v_2}, \text{ since } p_1v_1 = p_2v_2. \tag{145}$$

For average high-vacuum practice v_c = 3 per cent of the piston displacement. Assume $v_2 = 1$, $v_c = 0.03$, and $p_1 = 14.7$ pounds per square inch, and substitute these values in (145), thus:

$$\log_e \frac{14.7}{p_2} = 1 - \frac{0.03}{1.00}\,.$$

Whence $p_2 = 5.5$ pounds per square inch absolute, which corresponds to a vacuum of 18.6 inches of mercury.

Thus we see that the maximum load on the pump occurs when the vacuum is between 18 and 20 inches. If the vacuum is less than this, the load falls off because of the decreased difference in pressure. If the

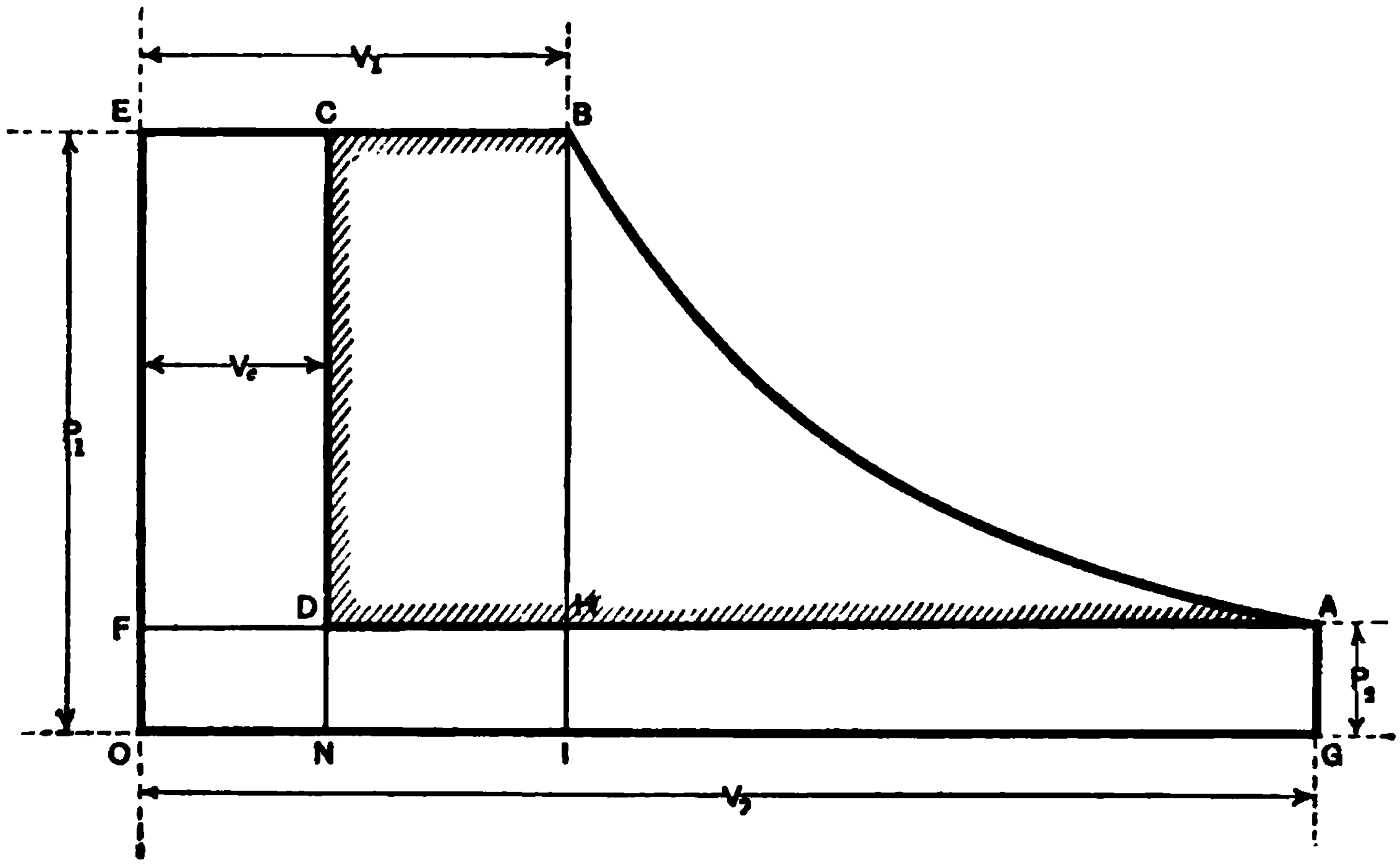


FIG. 290.

vacuum is greater, the load falls from the decrease in weight of air handled.

291a. Hot-well Pumps. — In high-vacuum surface condensers the condensed steam is often handled by a small independently driven pump called the *hot-well pump*. See paragraph 228.

Types of Air Pumps: Power & Engr., June 1, 1909, p. 963.

292. Centrifugal Pumps. — Centrifugal pumps consist of two essential elements, (1) a rotating impeller which draws in the water at its center, and (2) a stationary casting which guides the water thrown

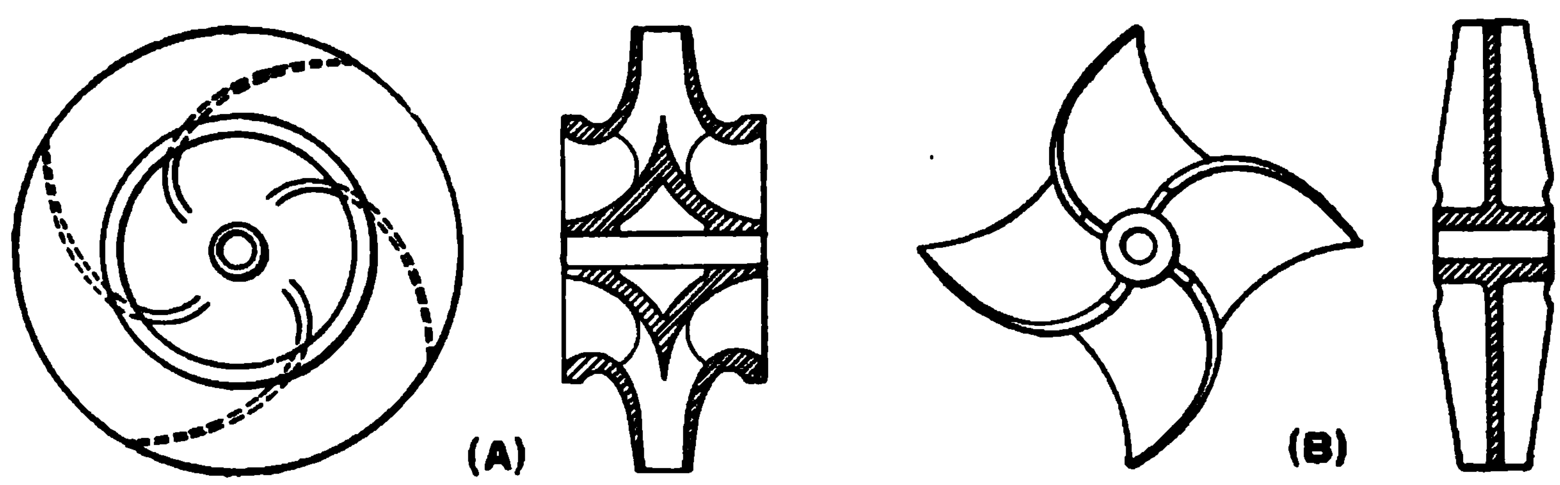


FIG. 291. Types of Impellers.

from the ends of the impeller to the discharge outlet. The impellers may be of the open type, Fig. 291(B), or closed, Fig. 291(A). The casing may be cylindrical and concentric with the impeller, Fig. 295, or of

spiral form, Fig. 292. The shape of the impeller and casing determines the efficiency of the pump and its adaptability to certain conditions of service.

Centrifugal pumps may be classified as

(1) Volute.
(2) Turbine.

293. Volute Pumps. — Fig. 292 shows a section through a typical volute pump. The casing is of spiral design, forming a gradually increasing water or "whirlpool" chamber for the purpose of partially converting velocity head to pressure head. The older forms of volute pumps were very inefficient, seldom delivering more than 50 per cent of the energy supplied, and usually not adapted to lifts greater than 50 feet. The modern pump gives efficiencies as high as 77 per cent, and lift is limited only by the speed of the impeller. As a general rule the volute pump is of single-stage construction, and best adapted to comparatively low lifts (under 80 feet), though an exception is found in the De Laval steam turbine driven volute pumps, which are made both single and multi-stage for lifts as high as 700 feet.

FIG. 292. A Typical Centrifugal Pump.

294. Turbine Pumps. — The directions of flow in the casing and from the impeller in a volute pump are at cross currents with each other, as shown in Fig. 293. The turbine pump, Fig. 294, is provided with a

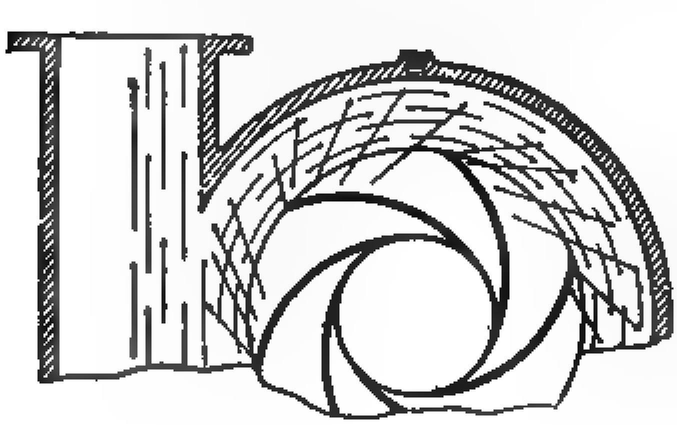


FIG. 293. Direction of Water from the Impellers of a Centrifugal Pump without Diffusion Vanes.

FIG. 294. Effect of Diffusion Vanes on the Direction of Water.

system of diffusion vanes or expanding ducts disposed between the periphery of the impeller and the annular casing somewhat like the guide vanes in a reaction turbine water wheel, so that the fluid emerges

tangentially at about the velocity in the casing. The casing is concentric with the impeller, since the diffusion vanes render the volute

FIG. 295. Lea-Degan Three-Stage Turbine Pump.

chamber unnecessary. For high lifts these pumps are compounded, thereby reducing the peripheral velocity and decreasing the friction losses. Fig. 295 shows a section through a three-stage turbine pump and Fig. 296 a section through a six-stage pump.

FIG. 296. Rateau Six-Stage Turbine Pump.

In view of past developments it is probable that the centrifugal pump will supplant to a considerable extent the present type of piston pump

for many uses. Efficiencies above 70 per cent are not unusual, and the head against which the pump may operate is limited only by the peripheral speed at which the impeller may be safely run. Some of the advantages of the modern high-grade centrifugal pump as compared with the piston type are:

(1) Low first cost,
(2) Compactness,
(3) Absence of valves and pistons,
(4) Low rate of depreciation,
(5) Uniform pressure and flow of water,
(6) Simplicity of design and ease of operation,
(7) Freedom from shock,
(8) High rotative speed, permitting direct connection to electric motors and steam turbines,
(9) Ability to handle dirty water, sewage, and the like,
(10) In case of stoppage of delivery, the pressure cannot increase beyond the predetermined working pressure, and
(11) Ease of repair.

Some of the disadvantages are:

(1) Efficiency not as high as the best grade of piston pumps,
(2) Cannot be direct connected to low-speed engines when high lifts are desired, and
(3) The rate of flow cannot be efficiently regulated for wide ranges in duty.

Theory of Centrifugal Pumps: Engr. U.S., Oct. 1, 1907, p. 908; Bulletin of Univ. of Wisconsin, No. 173; Trans. A.S.M.E., 54–470; Prac. Engr., Lond., Sept. 29, 1905; Jour. Asso. Eng. Soc., Dec., 1901; Centrifugal Pumps, Turbines, and Water Motors, C. H. Innes, Chap. XXVI.

Centrifugal Pumps, General Description: Trans. A.S.M.E., 26–764; Engr. U.S., Aug. 1, 1907, p. 723, Oct., 1907, p. 908, Oct. 15, 1907, p. 952; Power, March, 1907, p. 172; Eng. Mag., July, 1906; Elec. World, Jan. 12, 1907, p. 113; Eng. Rec., Feb. 9, 1907, p. 165; Machinery, Jan., 1907, p. 237, April, 1907, p. 442.

295. Performance of Centrifugal Pumps. — For best efficiency a centrifugal pump must be properly designed for the intended service as to curvature of vanes, diameter and speed of impeller, and number of stages. Figs. 298 to 300 are based upon experiments with De Laval centrifugal pumps. When a practically uniform head is required at constant speed with varying water supply as in city waterworks, hydraulic elevator systems, or boiler feeding, the impeller vanes are designed to give the characteristic curve illustrated in Fig. 298, which protects the motor from possible overload.

In dry-dock and other variable-head work, in order not to overload the motor, the power should be practically constant through wide variations of head and at the same time the efficiency should not vary seriously. A desirable characteristic for such a pump is illustrated in Fig. 299.

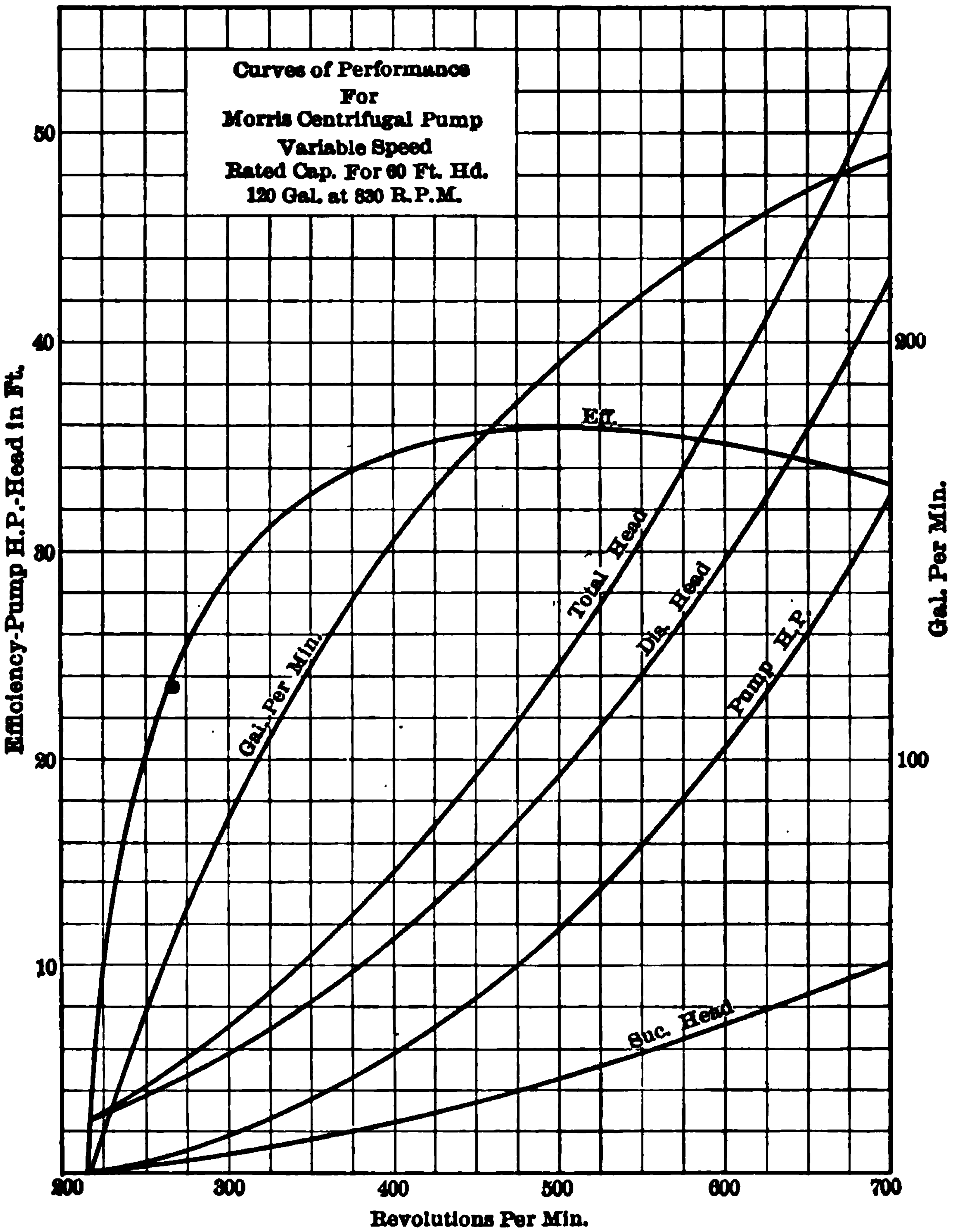


Fig. 297.

In water-supply systems in which the friction of the piping is a large part of the total head at full delivery, the characteristic shown in Fig. 300 is especially useful. Thus, when the system reduces its demand for water and the frictional head is consequently considerably reduced,

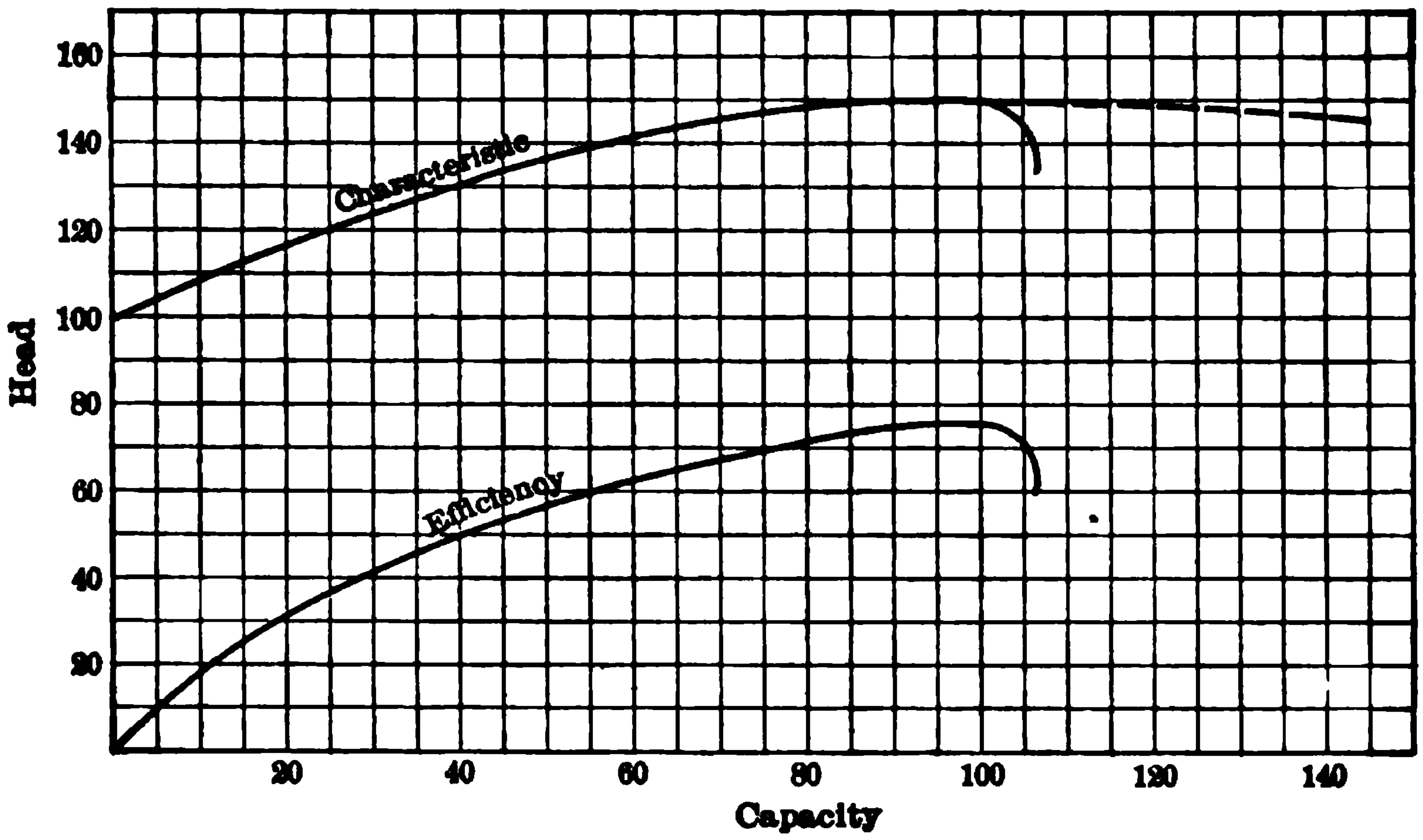


FIG. 298. Centrifugal Pump Characteristic for Hydraulic Elevator Service, Boiler Feeding, etc.

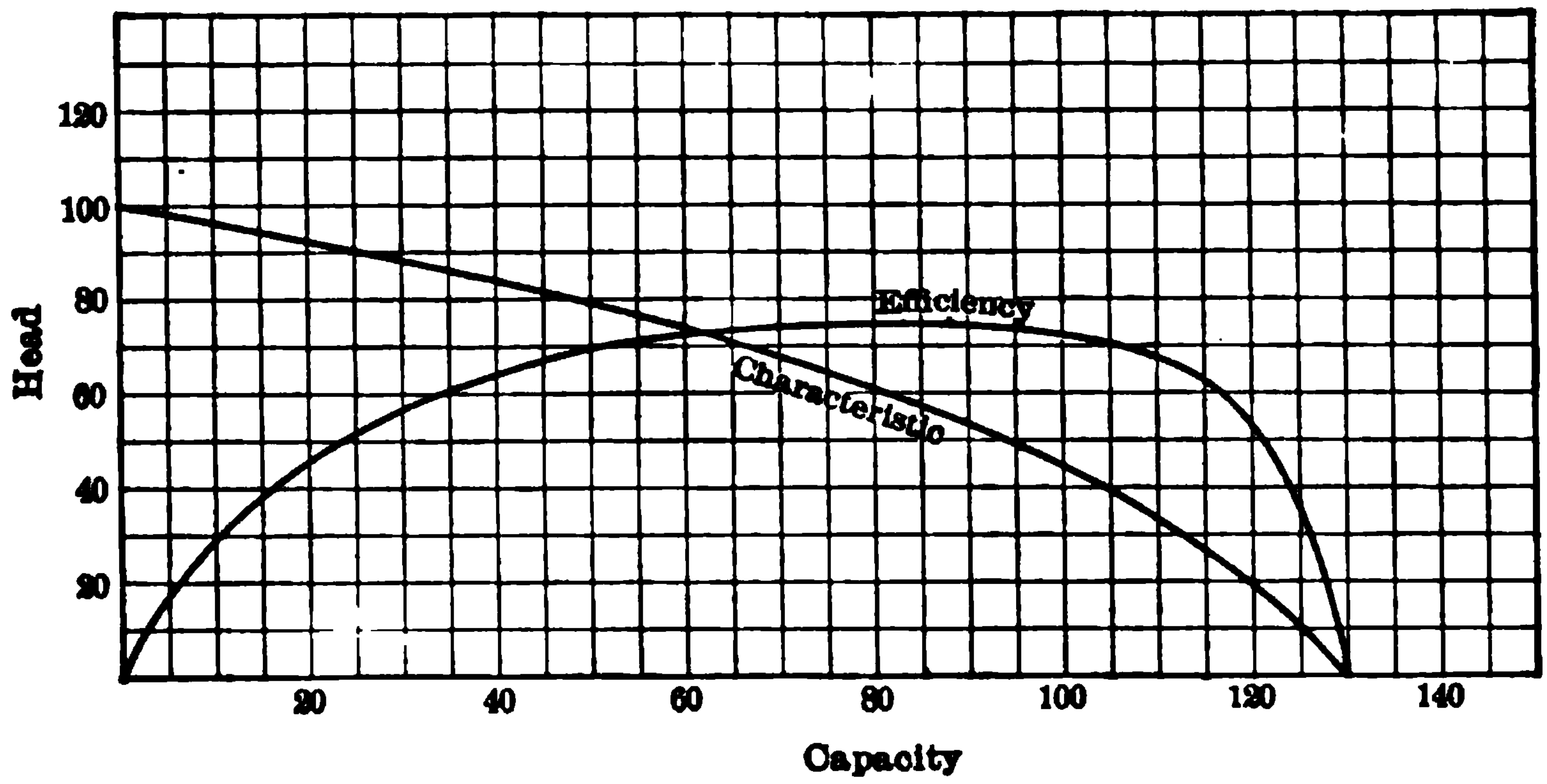


FIG. 299. Centrifugal Pump Characteristic for Dry-Dock Service.

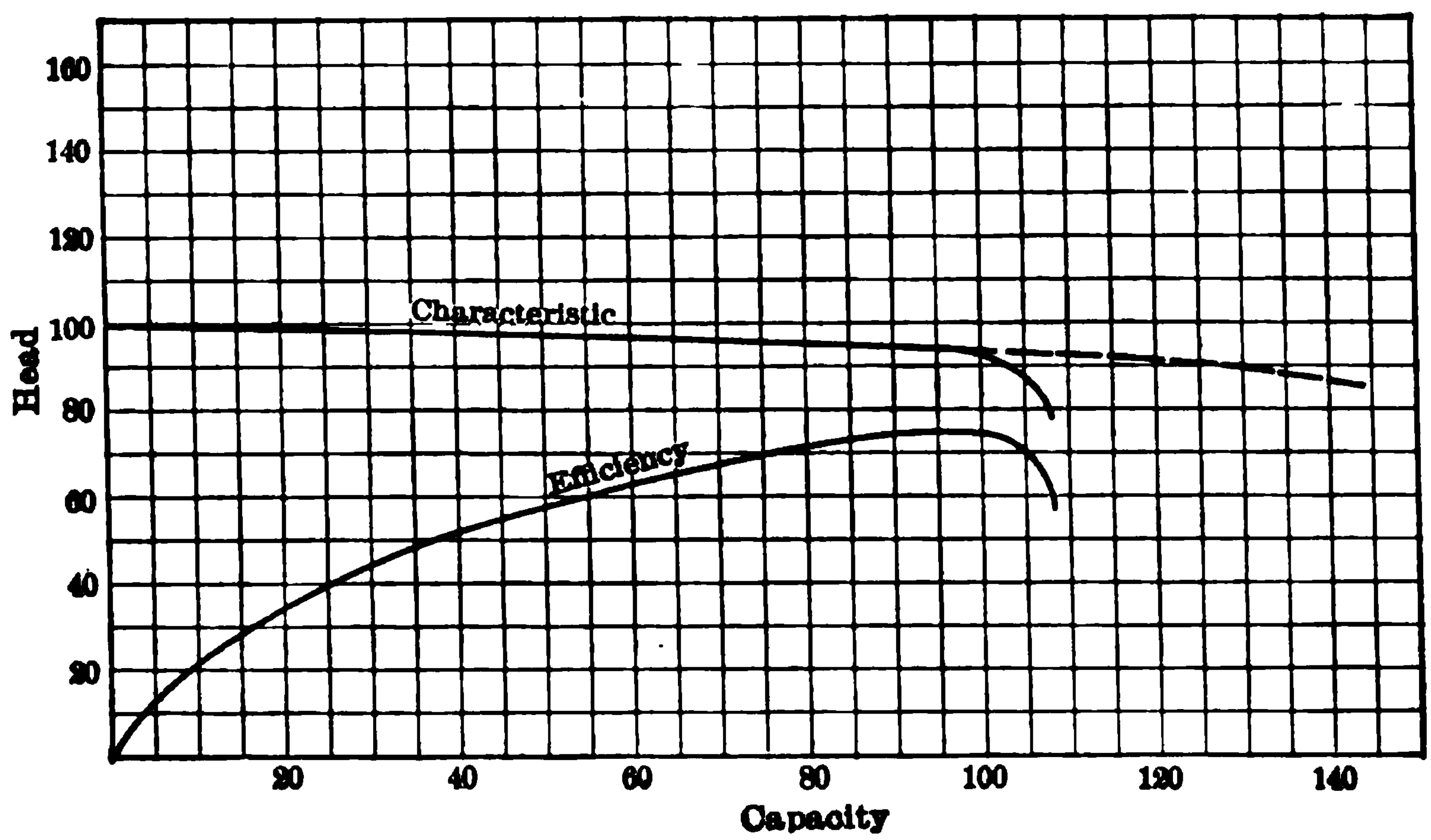


FIG. 300. Centrifugal Pump Characteristic for Water Works with Large Friction Head.

TABLE 73.

DATA PERTAINING TO SINGLE-STAGE CENTRIFUGAL PUMPS — CLASS "A," LAWRENCE PUMPS.

Diameter Discharge, Inches.	Diameter Suction, Inches.	Diameter Impeller, Inches.	Rated Capacity, Gals. per Min.	Velocity of Discharge, Ft. per Min.	Horse Power for Each Foot of Lift.	Total Head in Feet and Revolutions per Minute.										
						16 Ft.	20 Ft.	25 Ft.	30 Ft.	35 Ft.	40 Ft.	50 Ft.	60 Ft.	70 Ft.	80 Ft.	90 Ft.
1	1½	11	25	120	0.025	760	850	950	1040	1120	1200	1340	1470	1590	1695	1800
1½	2	14	55	120	0.042	620	690	765	830	890	950	1050	1140	1230	1315	1400
2	2½	18	100	120	0.063	460	510	570	620	670	710	795	870	940	1000	1060
3	3½	22	242	132	0.136	380	420	470	510	550	580	645	700	750	800	850
4	4½	26	430	132	0.217	310	340	370	405	435	465	515	560	600	640	670
5	6	29	734	144	0.309	279	295	320	350	370	400	435	470	510	545	570
6	6	32	1050	144	0.446	240	265	290	320	340	360	395	435	465	490	520
7	7	34	1439	144	0.606	220	250	275	300	320	340	375	405	435	465	490
8	8	36	1880	144	0.791	210	235	260	280	300	320	355	390	415	440	465
10	10	42	2938	144	1.265	190	210	230	247	265	280	310	335	360	380	400
12	12	45	4230	144	1.765	175	195	215	230	247	265	290	315	335	358	377
15	15	48	6610	144	2.775	165	185	200	220	235	250	275	295	317	335	355
18	18	52	9378	144	4.000	150	170	185	200	215	228	250	270	290	310	325

the pump would automatically adjust itself to the reduced head without change of speed. Figs. 297 to 304 are based upon experiment and show the relationship between speed, head, capacity, efficiency, and power consumption of various types of pumps.

Tables 73 and 74 give the capacity, speed, head, and power requirements for commercial sizes of centrifugal pumps, and may be used as a guide in selecting the size of pump for general service.

TABLE 74.

DATA PERTAINING TO WORTHINGTON MULTI-STAGE TURBINE PUMPS.

Diameter of Discharge Pipe, Inches.	Diameter of Suction Pipe, Inches.	Rated Capacity, Gallons per Minute.	* Horse Power per Foot of Lift.	Total Head in Feet, R.P.M., Number of Stages.							
				100 Ft.		200 Ft.		300 Ft.		400 Ft.	
				R.P.M.	Number of Stages.	R.P.M.	Number of Stages.	R.P.M.	Number of Stages.	R.P.M.	Number of Stages.
1	1.5	30	0.02	2000	2	......	....	......	....	......	......
1.5	2	45-60	0.0395	1500	2	2000	3	......	....	......	......
2	2.5	75-100	0.0625	1300	2	1800	3	1500	4	......	......
2.5	3	125-150	0.095	1200	2	1600	3	1300	4	......	5
3	4	200-250	0.134	1100	2	1400	3	1200	4	......	5
4	5	350-450	0.222	950	2	1200	3	1000	4	......	5
5	6	600-700	0.297	800	1	1300	2	1150	3	1050	4
6	7	800-1000	0.396	750	1	1200	2	1000	3	950	4
8	10	1500-1800	0.643	600	1	1000	2	800	3	780	4
10	12	2500-2800	1.00	500	1	800	2	700	3	670	4

* Horse power based on maximum capacity.

Tests of Centrifugal Pumps: Engr. U.S., Oct. 15, 1906, p. 685; Eng. and Min. Jour., April 14, 1906, p. 698; Eng. News, June 2, 1904, p. 512; Eng. Rec., July 1, 1905, p. 25, Sept. 29, 1906, p. 352; Iron Age, Sept. 1, 1904, p. 25; Machinery, Nov., 1906, p. 144; Power, Nov., 1906, p. 688; Trans. A.S.M.E., 22-262, 831; Jour. Am. Soc. Naval Engrs., 17-85.

298. Rotary Pumps. — Rotary pumps are often used for circulating cooling water in condenser installations, and give about the same efficiency as centrifugal pumps under similar conditions of operation. For moderate pressure and large volumes they offer the advantage of low rotative speed, thus permitting direct connection to slow-speed steam engines. At high speeds they are noisy, due chiefly to the gearing. They occupy considerably less space than piston pumps of the same capacity, but require more room than the centrifugal type.

Fig. 305 shows a section through a two-lobe cycloidal pump. The shafts are connected by wheel gearing, the power being applied to one of the shafts. The water is drawn in at I and forced out at O, the

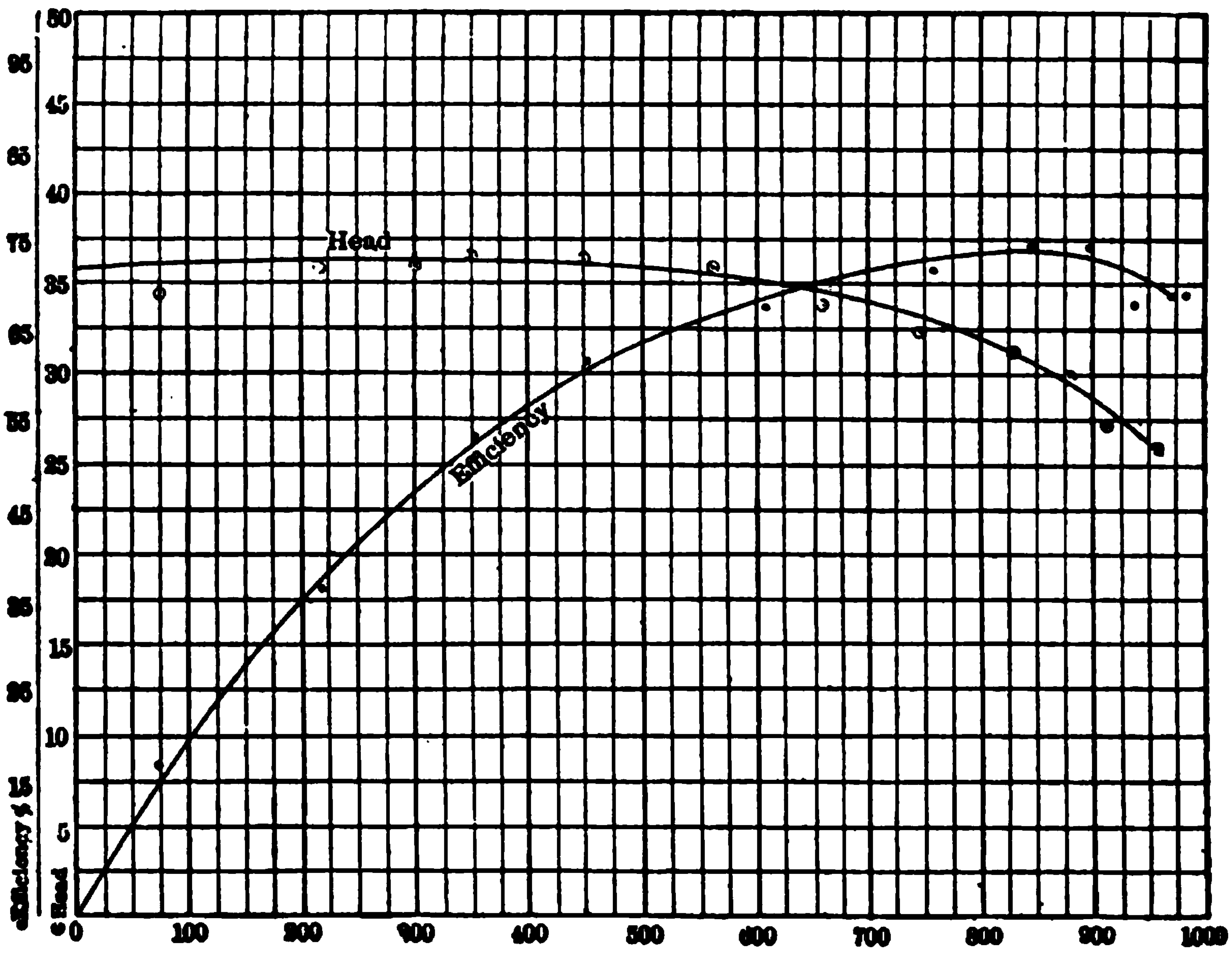


FIG. 301. Performance of a Six-Inch Worthington Conoidal Pump.

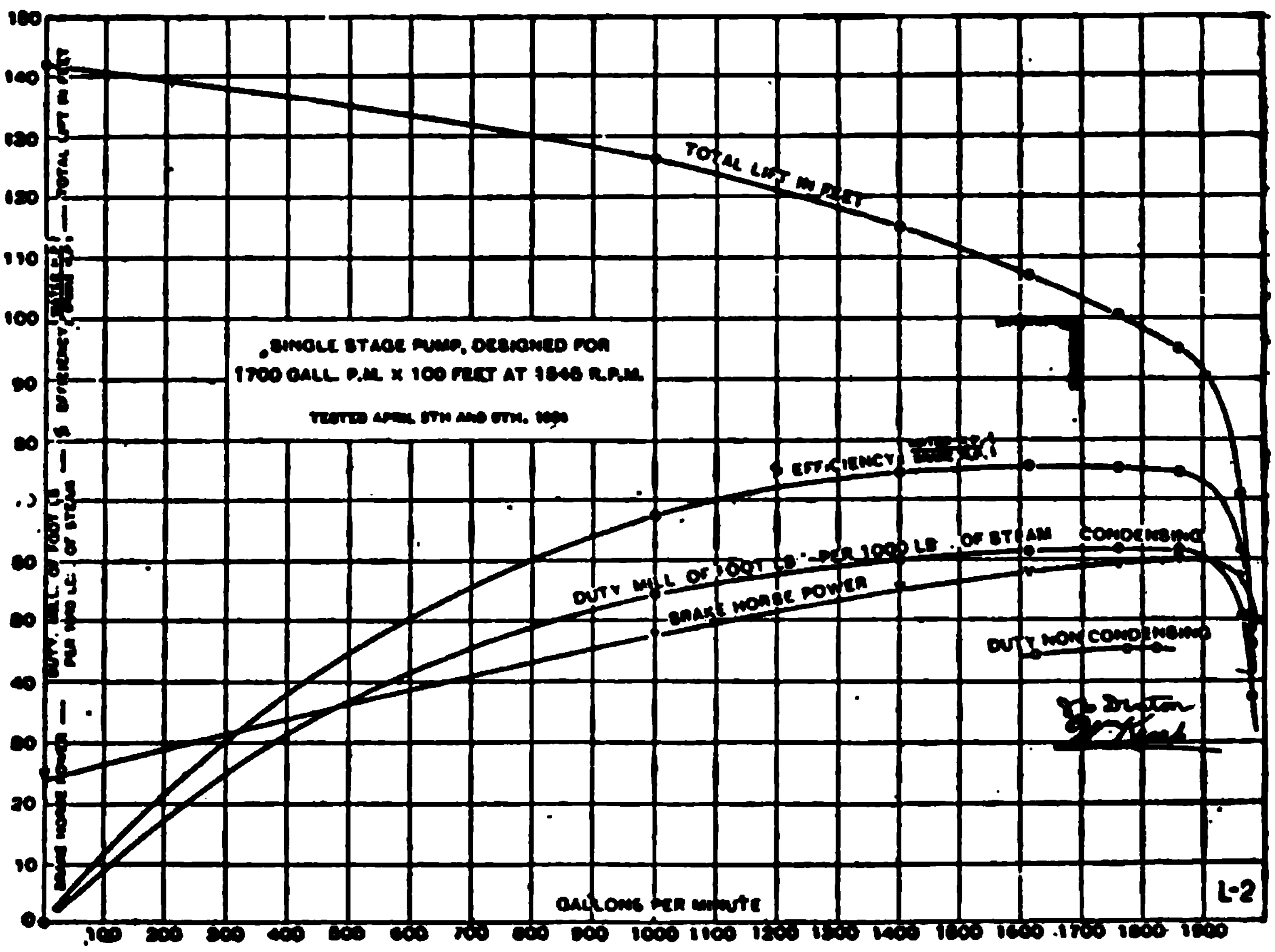


FIG. 302. Performance of a Single-Stage De Laval Centrifugal Pump.

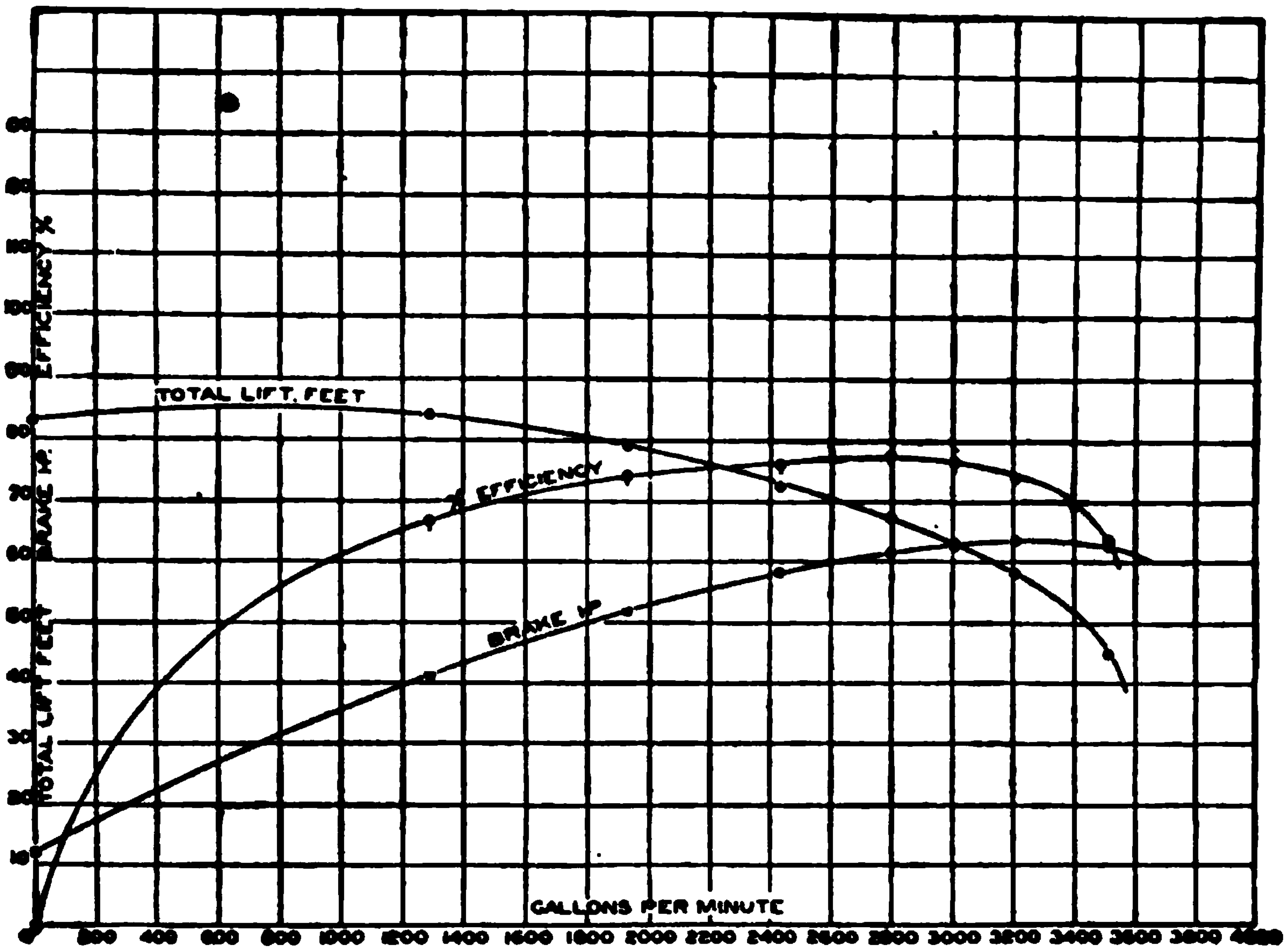


FIG. 303. Performance of a Two-Stage Lea-Degan Turbine Pump.

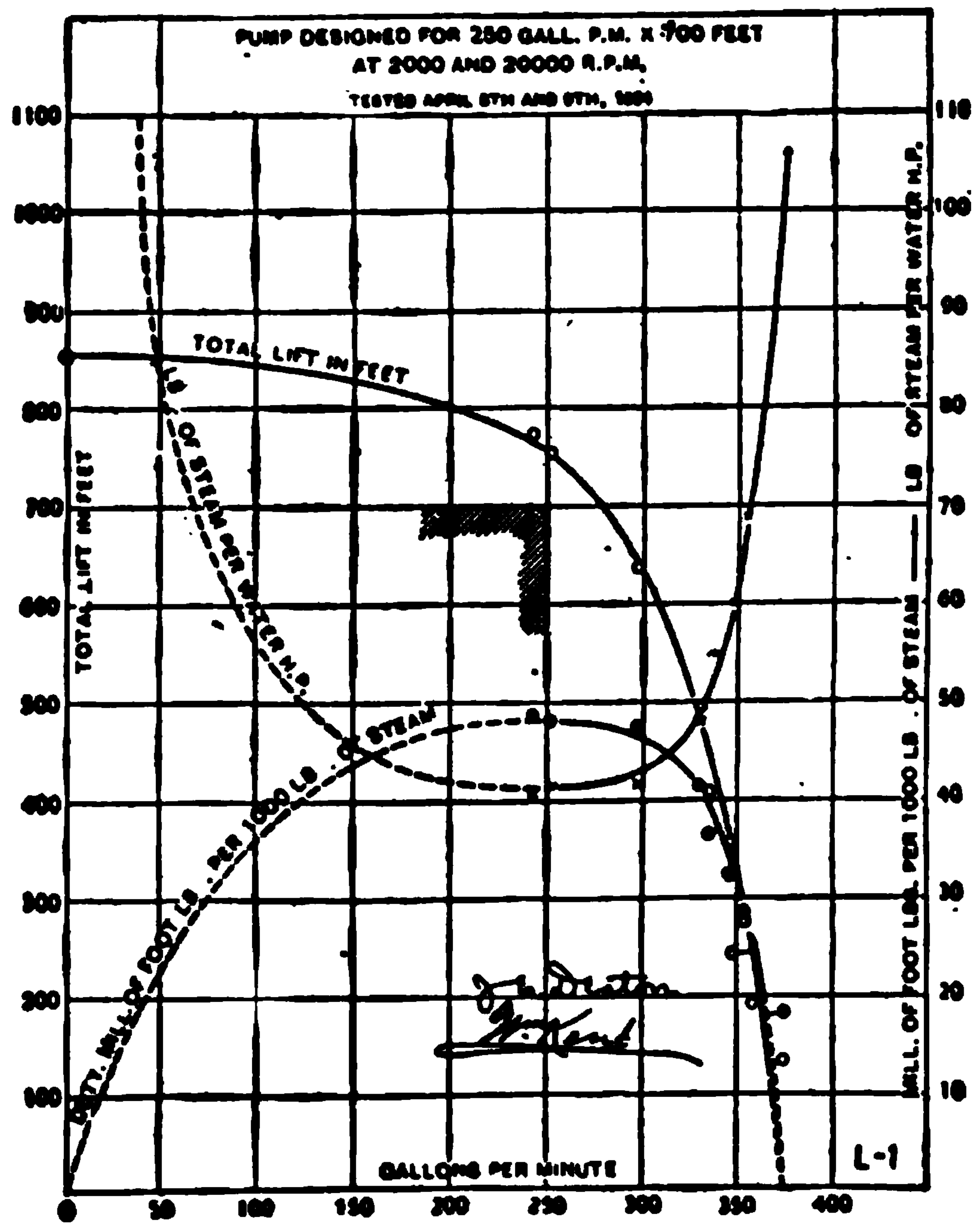


FIG. 304. Performance of a Two-Stage De Laval Centrifugal Pump.

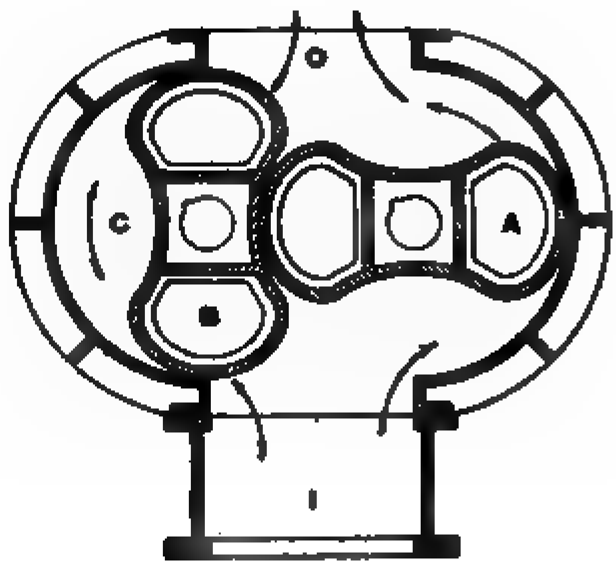
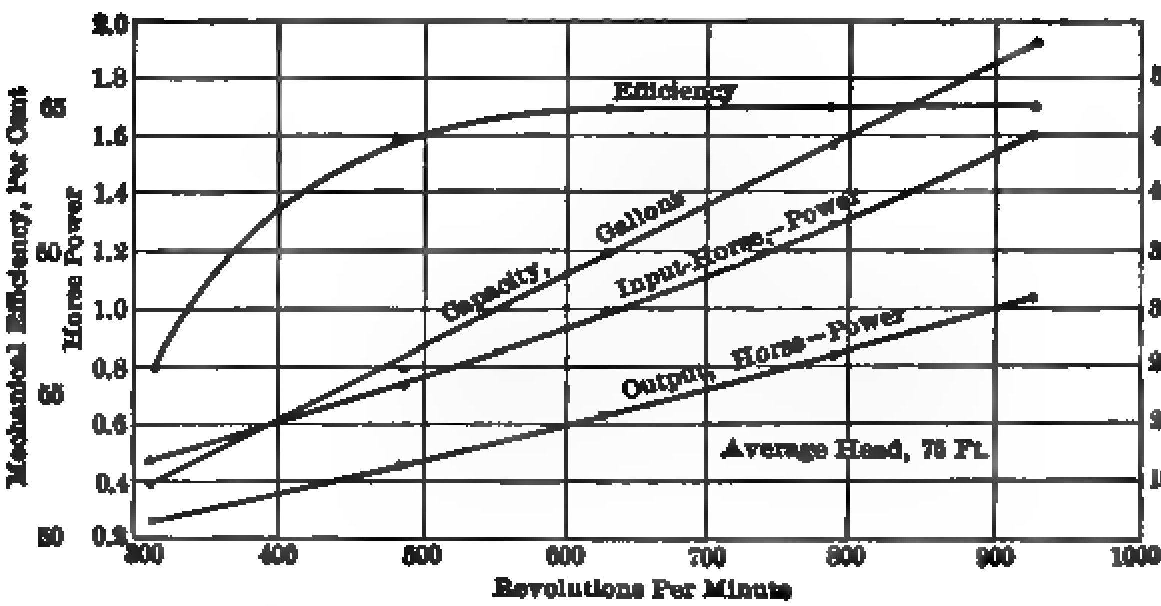
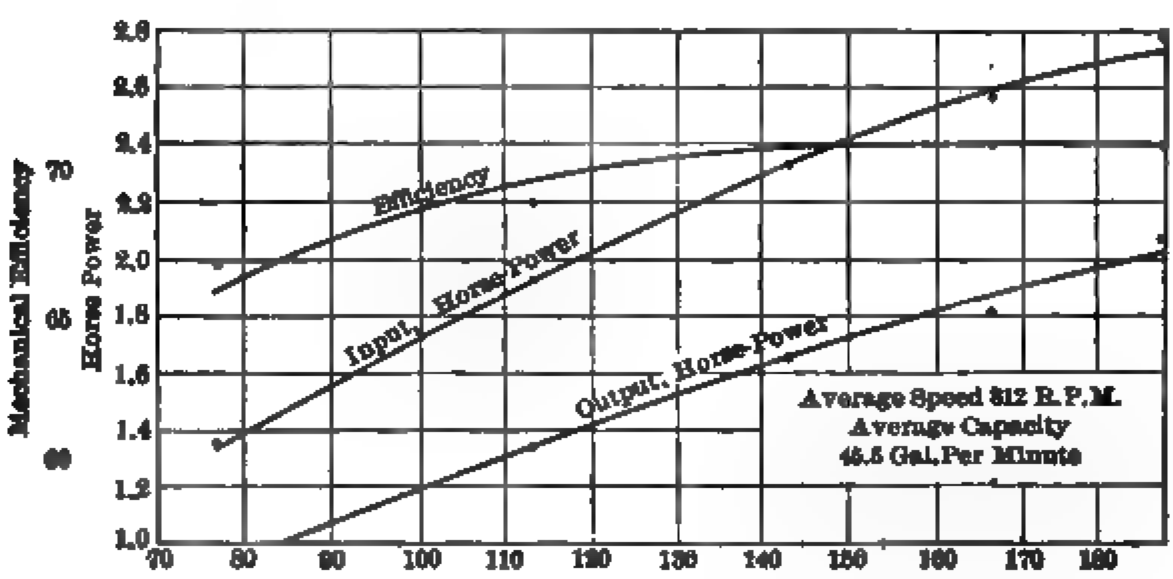


FIG. 305. Two-Lobe Cycloidal Pump.

FIG. 306. Rotary Pump with Movable Butment.



Head Constant, Speed Variable.



Speed Constant, Head Variable.

Fig. 307. Performance of a Small Rotary Pump.

displacement per revolution being equal to four times the volume of chamber *A*. There is no rubbing between impellers and casing. In this type of pump the pressure is independent of the speed of rotation, and the capacity varies almost directly with the speed. The slip varies from 5 to 20 per cent according to the discharge pressure.

Fig. 306 shows a section through a rotary pump with movable butment. Fig. 307 illustrates the performance of a 45-mm. Siemens-Schuckert rotary pump at different speeds and discharge pressures. (*Zeit. d. Ver. Deut. Ing.*, June 24, 1905, p. 1040.) Large rotary pumps give much higher efficiencies, but the general characteristics are about the same. A combined efficiency of pump and engine as high as 84 per cent has been recorded. (Trans. A.S.M.E., Vol. 24, p. 385.)

Fig. 308. 10,000,000 Gallon Circulating Pump.

Rotary Pumps: Engr. U.S., Jan. 1, 1904, p. 51; Am. Mach., March 12, 1896, p. 238, Jan. 25, 1906, p. 103; Trans. A.S.M.E., 24–385, Eng. News, March 1, 1900; p. 152; Power, Dec., 1903, p. 59, Aug., 1905, p. 477; The Constructor, Releaux, p. 226; Eng. Rec., Jan. 10, 1903, p. 59.

Tests of Rotary Pumps: Zeit. d. Ver. Deut. Ing., June 24, 1905, p. 1040; Trans. A.S.M.E., 28–503.

297. Circulating Pumps. — This term is ordinarily applied to the pumps which supply injection water to the condenser.

These types are found in practice: the piston, the centrifugal, and the rotary pump. Figs. 224 and 236 show the application of reciprocating pumps to condenser installations and Fig. 211a and Fig. 443 a similar application of centrifugal pumps.

For large volumes of water and low heads the centrifugal or rotary pump is generally adopted on account of minimum spacé requirements and low first cost.

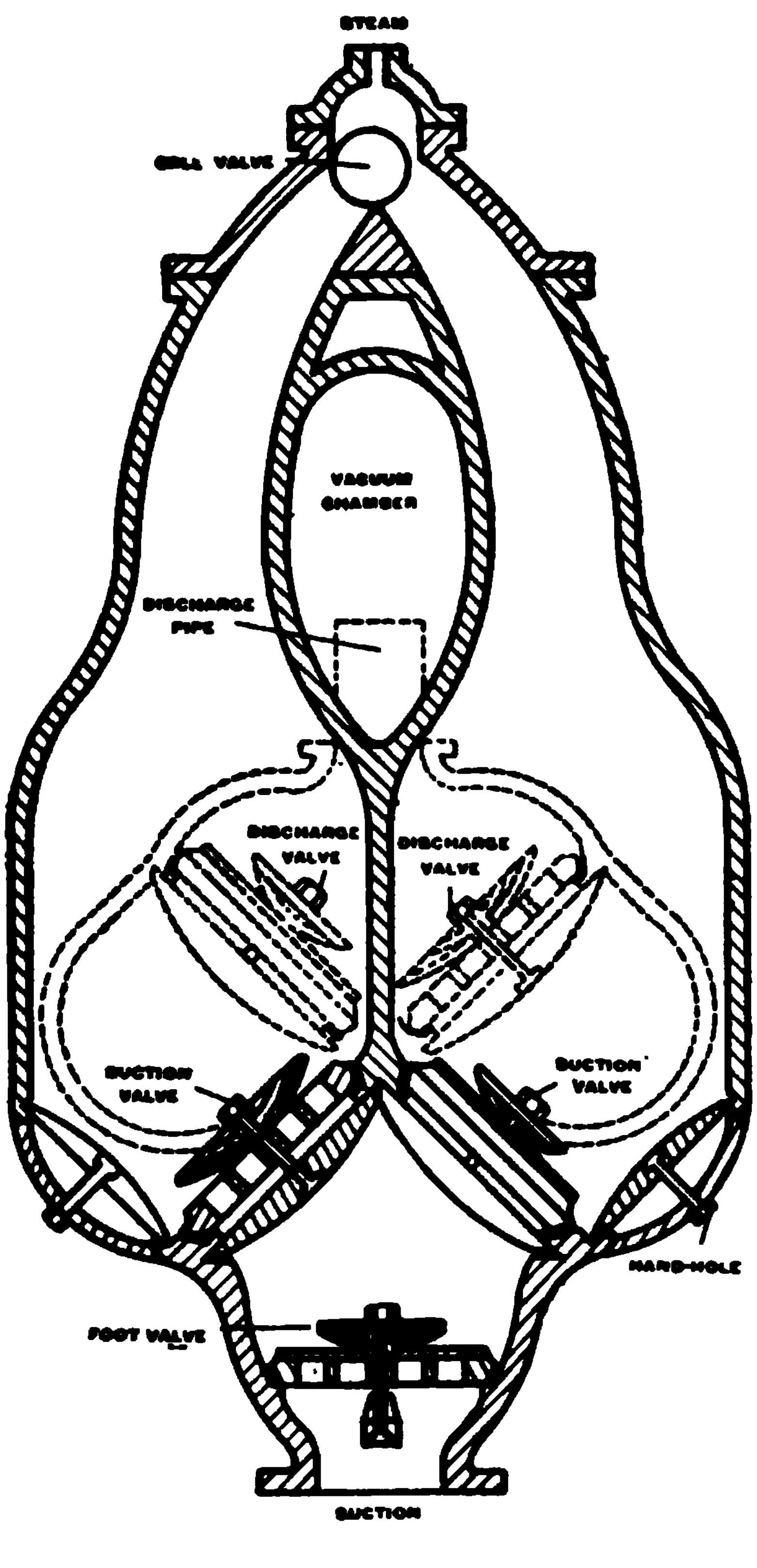


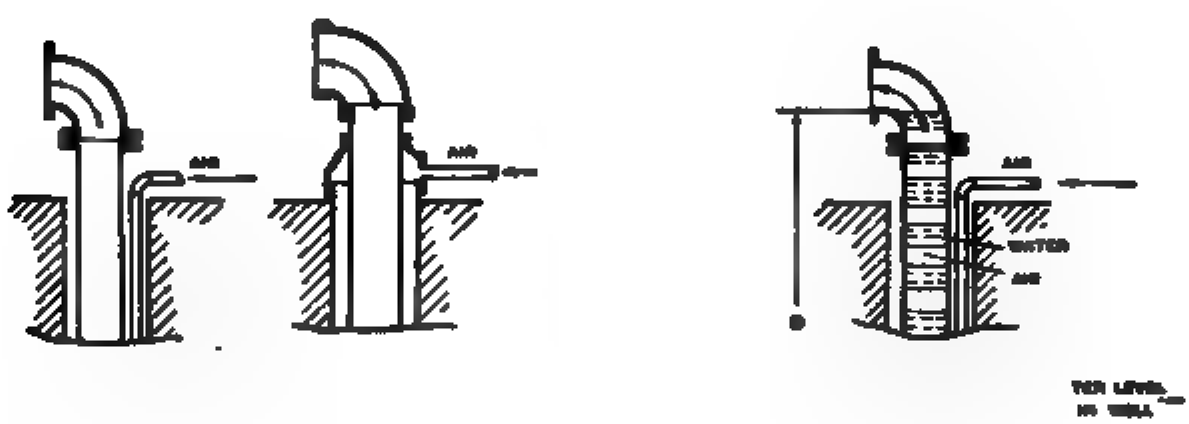
Fig. 309. The Pulsometer.

In very large central stations where the demand for circulating water is enormous and the lift is moderately high, the high-duty pumping engine is often installed. Fig. 308 shows a section through one of the nine high-duty circulating pumps at the New York Rapid Transit Company's power house. The steam end is operated by Corliss cylinders and is of the cross compound type. The maximum capacity is 10,000,000 gallons per day (24 hours) against a head of 50 feet at mean low water. The actual lift is much less than this, as the discharge is aided by the vacuum in the condenser.

Pumping Engines: Engr. U.S., Dec. 1, 1905, p. 786; Engng., Jan. 27, 1905, p. 132; Eng. Mag., Jan., 1905, p. 77; Eng. Rec., July 8, 1905, p. 58; Eng. News, Dec. 6, 1905, May 26, 1904; Trans. A.S.M.E., 3–141, 9–476, 12, 534, 975, 13, 83, 176, 14–1340, 15–1103, 16, 49, 169, 21, 327, 788, 1018.

298. Air Lift. — The air lift is a simple arrangement of piping whereby water may be raised by means of compressed air. There are no working parts, and no valves are employed except to regulate the supply of air. Its particular field of application lies in pumping water from a number of scattered wells, and on account of the total absence of working parts it is peculiarly adapted to handling water containing sand, grit, and the like The device consists of a partially submerged

water pipe and an air-supply pipe variously arranged as in Fig. 310(A) to 310(D). Compressed air forced into the water pipe at or near the bottom forms a series of bubbles or "pistons," as shown in *D*, which displace an equal volume of water. (For the theory of the air lift see *Compressed Air*, October, 1905, p. 3696.) The pressure required to operate the lift after it is once started is considerably less than the



(A) (B) (C) (D)

FIG. 310. Different Arrangements of the "Air Lift."

pressure of water due to the head, while that required to start it is slightly greater, consequently the pressure may be reduced after the pump begins to work properly. The successful operation of this device depends upon the ratio of the depth of submersion *A*, Fig. 310(D), to the total lift *B*, and the ratio of the area of the air pipe to that of the water pipe. The best results are obtained, in a general sense, when the ratio $\frac{A}{B}$ lies between 0.55 and 0.85, an average figure being 0.65, and when the area of the air pipe is 0.16 that of the water pipe.

The quantity of air needed may be closely approximated by the following equation (*Engineer*, London, Aug. 14, 1903, p. 173):

$$V = \frac{QL}{20},$$

in which

V = cubic feet of free air per minute.
Q = cubic feet of water per minute.
L = lift in feet above the surface of the water.

The velocity of the air should not exceed 4000 feet per minute.

The efficiency ("water" horse power divided by "air" horse power) varies from 30 to 50 per cent, increasing as the ratio $\frac{A}{B}$ increases from 0.55 to 0.85. (*Engineer*, U.S., Aug. 15, 1904, p. 564.) A number of tests give efficiencies (" water " horse power divided by I.H.P. of steam cylinder) varying from 20 to 40 per cent. The horse power required to compress one cubic foot of free air to different pressures per square inch, as determined from actual practice, is approximately as follows:

Pressure in Pounds.	Horse Power Required to Compress 1 Cubic Foot.	Pressure in Pounds.	Horse Power Required to Compress 1 Cubic Foot.
176	0.434	60	0.159
140	0.376	45	0.145
100	0.201	30	0.121
80	0.189		

(Engr., Lond., Aug. 14, 1903, p. 174, Dec. 11, 1903, p. 568, Feb. 12, 1904, p. 172.)

When it becomes necessary to raise water to a height exceeding say 175 feet above the level in the well, it is customary to use two or more pumps, the total lift being divided between them.

Air Lift: Engr., Lond., Aug. 14, 1903, p. 173, Dec. 11, 1903, p. 568, Feb. 12, 1904, p. 172, Jan. 10, 1908; Eng. Rec., Jan. 7, 1905, p. 8; Engineering, Jan. 22, 1904, p. 135, Jan. 29, 1904, p. 166; Compressed Air, Aug., 1903, Oct., 1905; Engr. U.S., July 15, 1903, p. 547, Sept. 21, 1906, Aug. 15, 1904, p. 564; Mech. Engr., Aug. 20, 1904; Engng. Rev., July, 1904; Power, March, 1905, p. 173; Eng. News, Jan. 16, 1908.

Pulsometer: Tech. Quar., Sept., 1901; Public Works, Aug. 15, 1904; Engr. U.S., July 15, 1904; Experimental Eng., Carpenter, p. 621; Thermodynamics, Wood, p. 293; Trans. A.S.M.E., 13–211.

Cost of Operating American Pumping Stations: Eng. Rec., Aug. 6, 1904; Proc. Engrs. Club of Phil., Oct., 1906.

Complete Description of Various American Types of Steam, Rotary, and Centrifugal Pumps: Engr. U.S., Jan. 1, 1904.

The Growth of the Pumping Station: Eng. Rec., July 14, 1906, p. 50.

The Selection of Waterworks Pumping Machinery: Eng. News, Vol. 52, p. 39.

Centrifugal Pump for Boiler Feeding: Power & Engr., Mar. 15, 1910.

Recent Records of High Duty Pumping Engine: Eng. News, Feb. 3, 1910.

CHAPTER XIV.

SEPARATORS, TRAPS, DRAINS.

299. Live-Steam Separators. General. — The function of a steam separator is the removal of entrained water from steam.

Unless a boiler is liberally provided with superheating surface, the steam may contain an amount of moisture varying from 0.3 to 5 per cent. If the boiler is poorly proportioned or forced far above its rating, this percentage may be greatly increased. The quality of the steam is still further reduced by condensation in the steam pipe, which may vary from one to ten per cent, depending upon the length of pipe and efficiency of covering.

One of the effects of moisture in steam is to increase its density and reduce its elastic force. It also increases its conductivity, so that during the work of expansion more heat is absorbed from the walls of the cylinder and discharged into the atmosphere or into the condenser without doing useful work. (Ewing, "The Steam Engine," p. 151.) Although the heat loss from this cause is small, the danger arising from the introduction of a considerable amount of water in the cylinder renders the removal of the moisture necessary. See page 248 for influence of moisture on steam consumption.

The essentials of a good separator are high efficiency as a water eliminator, ample storage capacity for any sudden influx of water, simplicity and durability in construction, and small resistance to the current of steam passing through. A good separator may be relied upon to remove practically all of the moisture from steam containing under ten per cent entrainment and all but two per cent from steam containing as much as twenty per cent. (*Engineer*, U.S., Jan. 15, 1904.)

Table 75 gives the results of a series of tests made by Professor R. C. Carpenter in 1891 of six steam separators. (*Power*, July, 1891, p. 9.) Conclusions from these tests were:

1. That no relation existed between the volume of the several separators and their efficiency.

2. No marked decrease in pressure was shown by any of the separators, the most being 1.7 pounds by separator *E*.

3. Although changed direction, reduced velocity, and perhaps centrifugal force are necessary for good separation, still some means must be provided to lead the water out of the current of the steam.

A series of tests made at Armour Institute of Technology in 1905 on a number of separators showed that the *efficiency of separation decreased as the velocity of the steam increased.** At the low velocity of 500 feet per minute all separators were equally efficient, at a velocity of 5000 feet per minute several had little effect on eliminating the moisture present, and at a velocity of 8000 feet per minute only one gave efficient results.

TABLE 75.

TESTS OF STEAM SEPARATORS.

(R. C. Carpenter.)

Make of Separator.	Test with Steam of about 10 per Cent of Moisture.			Tests with Varying Moisture.		
	Quality of Steam Before.	Quality of Steam After.	Efficiency.	Quality of Steam Before.	Quality of Steam After.	Average Efficiency.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
B	87.0	98.8	90.8	66.1–97.5	97.8–99	87.6
A	90.1	98.0	80.0	51.9–98	97.9–99.1	76.4
D	89.6	95.8	59.6	72.2–96.1	95.5–98.2	71.7
C	90.6	93.7	33.0	67.1–96.8	93.7–98.4	63.4
E	88.4	90.2	15.5	68.6–98.1	79.3–98.5	36.9
F	88.9	92.1	28.8	70.4–97.7	84.1–97.9	28.4

300. Classification of Separators. — Separators are based on one or more of the following principles of action:

1. *Reverse current.* The direction of the flow is abruptly changed, usually through 180 degrees. This causes the water in the steam, on account of its greater specific gravity, to be thrown into a receiving vessel, while the steam passes on in a reverse direction.

2. *Centrifugal force.* A rotary motion is imparted to the steam whereby entrained water particles are eliminated by centrifugal force.

3. *Baffle plates.* The flow is interrupted by corrugated or fluted plates to the surfaces of which the water particles adhere and from which they fall by gravity to the well below.

4. *Mesh.* The separation is brought about by mechanical filtration through screens or meshes.

The following outline shows the classification of typical separators, in accordance with the above principles:

* See Power, May 11, 1909, p. 834.

Live-steam separators	Reverse current	Hoppes, Fig. 311.
		Stratton, Fig. 312.
	Centrifugal	Keystone, Fig. 313.
		Mosher.
		Robertson.
	Baffle plate	Bundy, Fig. 314.
		Austin, Fig. 315.
		Detroit.
	Mesh	Direct, Fig. 316.
		Potter mesh.
Exhaust-steam separators	Jacketed baffle	Baum, Fig. 317.
	Absorption	Loew, Fig. 318.

301. Reverse-Current Steam Separators. — Fig. 311 shows a section through a Hoppes steam separator and illustrates the principle of reverse-current separation. Steam may flow through in either direction. Both the inlet and outlet ports are surrounded by gutters C, C, partly filled with water, which intercept the moisture following the surface of the pipe, while the downward plunge of the steam throws the entrained water to the bottom of the separator. The condensation is carried from the troughs by pipe P to the well below, from which it is trapped at D in the usual way. The velocity of the steam in passing through this separator is greatly reduced to prevent the steam from taking up the water in the bottom of the well. This is brought about by increasing the area of the passage through the separator.

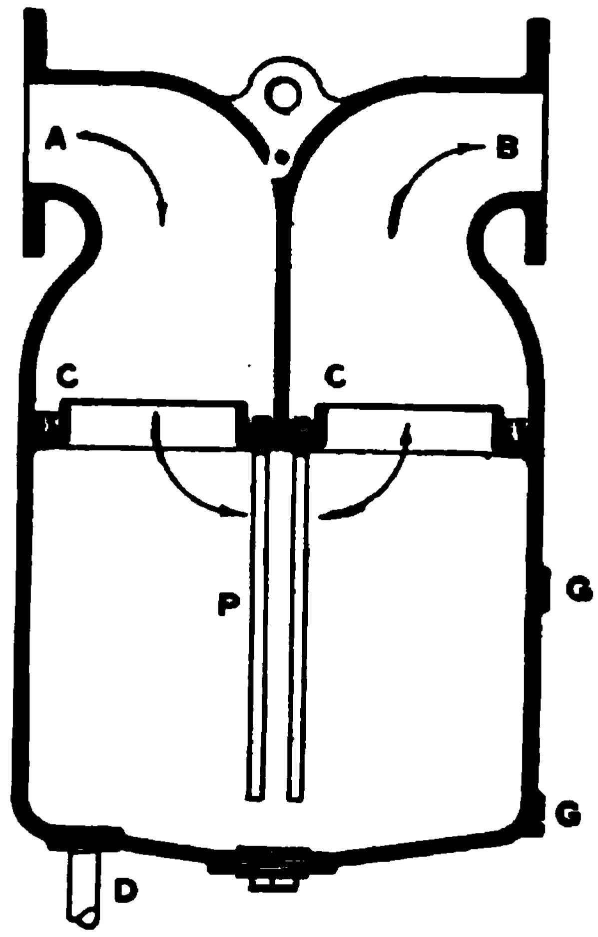


FIG. 311. Hoppes Steam Separator.

Fig. 312 gives a sectional view of a Stratton separator, which, though primarily of the reverse-current type, embodies also the principle of centrifugal force. The separator consists of a vertical cast-iron cylinder with an internal central pipe C extending from the top downward for about half the height of the apparatus, leaving an annular space between the two. The current of steam on entering is deflected by a curved partition and thrown tangentially to the annular space at the side, near to top of the apparatus. It is thus whirled around with all the velocity of influx, producing the centrifugal action which throws the particles of water against the outer cylinder. These adhere to the surface, so that the water runs down continuously in a thin sheet around

the outer shell into the receptacle below. The steam, following in a spiral course to the bottom of the internal pipe, abruptly enters it, and passes upward and out of the separator without having once crossed the stream of separated water. The rapid rotation of the current of steam imparts a whirling motion to the separated water which tends to interfere with its proper discharge from the apparatus. The separator has therefore been provided with wings or ribs *E* projecting at an acute angle to the course of the current, which have the effect of breaking up this whirling motion and allowing the water to settle quietly at the bottom, whence it passes off through the drain pipe *D*.

FIG. 312. Stratton Steam Separator. FIG. 313. Keystone Steam Separator.

302. Centrifugal Steam Separators. — Fig. 313 shows a section through a Keystone or Simpson's centrifugal separator. The separator consists of a cast-iron cylinder with vertical pipe *C* extending downward about two-thirds of the whole length; this pipe has a thread or screw wound spirally around it, the space between the threads being somewhat greater than the area of the steam pipe. The steam passing around the spiral course causes the water to be thrown against the outer walls by centrifugal force, while the dry steam passes through

the small holes in the central pipe. The water passes down the outer walls, where its motion is arrested by obstructing ribs E, and is thence carried away by a drip pipe D to a suitable drain.

303. Baffle-Plate Steam Separators. — Fig. 314 gives an interior view of a Bundy separator and illustrates the application of baffle plates for live steam separation. This separator consists of a rectangular cast-iron casing with a cylindrical receiver beneath it. Directly across the steam passage are baffle plates corrugated for the reception of entrained water. The plates consist of vertical castings, each containing a main artery or channel which leads directly to the receiver. The fronts of the plates are flat, with a series of recesses sloping inwards and downwards, terminating in an opening of capillary size leading to the main artery. The plates are staggered, so that the steam must impinge against all of them in its passage. The particles of water adhere to the plates, collect, and fall by gravity into the receiver. The flanges at the bottom constrict the opening of the reservoir so as to prevent the steam from picking up any portion of the water.

Fig. 314. Bundy Steam Separator.

Fig. 315 shows a section through an Austin separator and illustrates another class embodying the fluted baffle plate principle. The steam in passing through the chamber impinges against the fluted baffle plate B. The moisture adheres to the surfaces, collects, and trickles along the corrugations to the bottom of the well. These corrugations are formed in such a manner that the steam cannot come in contact with the water particles after they have been once eliminated. A perforated diaphragm D prevents the water in the well from coming in contact with the steam. The current of steam is also reversed, thus giving additional separating properties to the apparatus.

Fig. 315. Austin Steam Separator.

304. Mesh Separators. — Fig. 316 shows a section through a "direct" separator, illustrating the principle of mesh separation. These separa-

tors are made with steel bodies and cast-iron heads and bases, in all sizes up to six inches inclusive, the larger sizes being constructed of cast iron or boiler plate. The cone C, perforated lining E, and diaphragm S are made of cold-rolled copper; the cone O is a substantial gray-iron casting, resting on three cast-iron supports hooked over the top of inner pipe as indicated. The method of operation is as follows: The accumulated moisture around the walls of the steam pipe is caught by the upper edge of cone C and carried down back of lining E to the water chamber. The current of steam entering the separator impinges upon the conical surface, which is composed of solid plate O covered with sieve S, through which water may freely pass but from which it cannot readily escape. Passing through the sieve and depositing on the solid surface of the cone O, this water is carried by conductors P to the water chamber. Perforated lining E permits the moisture content of the steam to pass through the opening to the water below and prevents it from coming in contact again with the current of steam. A trough is provided at the lower edge of the inverted cup which leads all the water that may adhere to it to the water chamber. The steam flows through the passages indicated by arrows and is subjected to a whipsnapping action which tends to throw off any remaining moisture. The perforated plate D prevents the steam from picking water out of the water chamber.

FIG. 316. "Direct" Steam Separator.

305. Location. — Live-steam separators may be located

1. Inside the boiler.
2. Between boiler and engine.
3. At the steam chest.

Where the steam pipe is very short, and particularly in marine and locomotive work where the tossing of the boiler induces excessive priming, the separator may be placed inside the boiler and its function becomes that of a dry pipe. In this location it prevents the water due to foaming and priming from passing to the engine, and

reduces condensation in the pipe by supplying dry steam. The "Potter mesh" and the "De Rycke centrifugal" are types of separators designed for this service.

The arrangement of separator between engine and boiler, other than at the throttle, or inside the boiler is sometimes necessary for economy of space. Where possible, however, the separator should be placed close to the steam chest.

Current practice recommends that a receiver separator, which is an ordinary separator with a volume of two to four times that of the high-pressure cylinder, be placed close to the engine if the load is intermittent or sharply fluctuating. This forms a cushion for absorbing the force of the blows caused by cut-off, delivers steam at a practically uniform pressure, and reduces the vibration of the piping to a minimum. It also provides a reservoir for sudden demands made by the engine. Smaller pipes and higher velocities may be used with this arrangement.

306. Exhaust-Steam Separators and Oil Eliminators. — The function of an exhaust-steam separator is the removal of cylinder oil from the steam exhausted by engines and pumps. In plants where exhaust steam is used for heating it is quite essential to remove the oil from the steam before it enters the heating system, for the oil not only reduces the efficiency of the radiators by coating them with an excellent non-conducting film but is an element of danger to the boiler itself. In condensing plants the separator will prevent the oil from fouling the condenser tubes and those of the vacuum heater if one is installed; this is an important factor, since the oil or grease lowers the efficiency of the heat transmission.

In a general sense a live-steam separator is also an oil eliminator, and all the separators previously described perform this function to a certain extent, since the underlying principles governing the elimination of oil from exhaust steam are similar to those employed in removing water from steam. Most of the separators described above are also designed, in lighter form, as oil eliminators, but by far the greater number are based on the fluted baffle plate principle, of which the Hine, Bundy, Cochrane, Utility, Peerless, and Keiley are well-known examples. This type of oil separator will eliminate a considerable portion of the oil in the steam provided the baffle plates or corrugated surfaces are frequently cleaned.

The following is taken from the report of Professor R. Burnham of the Armour Institute of Technology on the test of a six-inch horizontal oil separator of the baffle-plate type:

"For purposes of test the separator was placed in the exhaust line

of a 9 x 18 x 24 cross compound Corliss engine running under its maximum load at 80 pounds pressure and exhausting into a Wheeler surface condenser against 26 inches vacuum.

"Cylinder oil was fed through the lubricators of the high and low pressure cylinders at the rate of from 5 to 20 drops per minute, a record being made of the exact quantity of oil fed per hour. The separator was so arranged, by means of a receiver connected to the air pump, that the accumulation of oil and water could be readily trapped from it at any time. In order to determine the quantity of oil given up by the condenser, and not properly charged against the separator, each series of efficiency tests was preceded by a run of three hours during which time no oil whatever was fed to the cylinders. During the last hour a record was made of the weight of steam used and a sample of the condenser discharge retained for analysis.

"The efficiency tests were made by feeding at an excessive rate through the lubricators as described above, and when conditions became practically constant, records were made for one hour, of the weight of oil used, weight of condensed steam, and drain from separator. Samples of the two latter were retained for analysis and the percentage of oil in them accurately determined, correction being made for the oil given up by the condenser. A second series of tests was made exhausting at atmospheric pressure. The results obtained are tabulated below.

	Exhausting into 26-inch Vacuum.			Exhausting at Atmospheric Pressure.	
Oil in condensed steam with no oil feeding. (Charged to condenser.) Pounds per hour	.051	.057	.0559	.0353	.0340
Oil fed to cylinder, pounds per hour.......	.401	.562	.934	.621	.710
Steam condensed per hour, pounds........	1000	1120	1096	905	872
Oil caught by separator, per hour, pounds A	.341	.450	.743	.552	.583
Oil in condensed steam (corrected), pounds per hour..........................B	.009	.010	.0096	.0071	.0050
Percentage of oil in condensed steam by weight, per cent........................	.0009	.001	.00088	.00078	.00057
Efficiency of separator, per cent $\frac{A}{A+B}$ ...	97.4	97.8	98.8	98.7	99.1

"There was practically no free oil on the surface of the condenser discharge in any case, the small quantity of oil which passed the separator (from 5 to 10 parts in a million of water by weight) existing as an emulsion, imparting a slight milky color to the water."

It is a well-established fact that oil can be more effectually removed from wet than from dry steam, and some makers, notably the Austin

Separator Company, inject a cold-water spray into the separator chamber. A similar result is brought about in the Baum separator, Fig. 317, in which the corrugated baffle plate is hollow and cold water is forced through the chamber thus formed. Referring to Fig. 317: The diverged baffle plate forms the wall of a chamber in which cold water is continually circulated. This circulation causes moisture to appear on the baffle plate surface. The particles of oil, coming in contact with this moist surface as the steam current is diverged, adhere to it and fall by gravity into the well below, where they are completely isolated from the purified steam. A large portion of the oil and water, however, does not enter the separator at all but is caught by the inside ledge near the junction of the exhaust pipe and the separator. The oil and condensation which are carried along the bottom of the pipe come in contact with this ledge and are carried directly to the outlet pipe.

Fig. 317. Baum Oil Separator.

A very successful method of removing oil from steam is to project the steam on to the surface of a body of water. The water may be hot or cold and will hold the oil if it once reaches the surface. It is essential, however, to reduce the velocity of the steam as it passes on its way to the outlet. Baldwin's grease separator is based upon this principle. (Baldwin on Heating, p. 234.)

The most efficient method of removing oil is by combined filtration and absorption. (*Engineering News*, May 22, 1902, p. 406.) A large chamber filled with coke, brick, broken tile, or other absorption material is placed in series with the exhaust pipe. The steam passing through this chamber is entirely freed from oil and moisture, provided the absorbing material is sufficient in quantity and is replenished as soon as it becomes saturated with oil. The annoyance attending the removal and replenishing of the absorbing material at frequent intervals and the great size of the apparatus are serious drawbacks. An example of this system of purification in which many of the objectionable features are reduced to a minimum is the Loew grease and oil separator, Fig. 318.

The exhaust steam enters the chamber at the top, strikes a large deflecting plate shaped like an inverted V, and permits part of the condensation and oil to be drawn off by the drain pipe. The steam then rises and is deflected, as indicated, against a series of shelves filled with fibrous material covered with coarse wire screens. The grease is removed from each shelf by suitable drains. This apparatus is sectional and any number of sections may be added without affecting the rest.

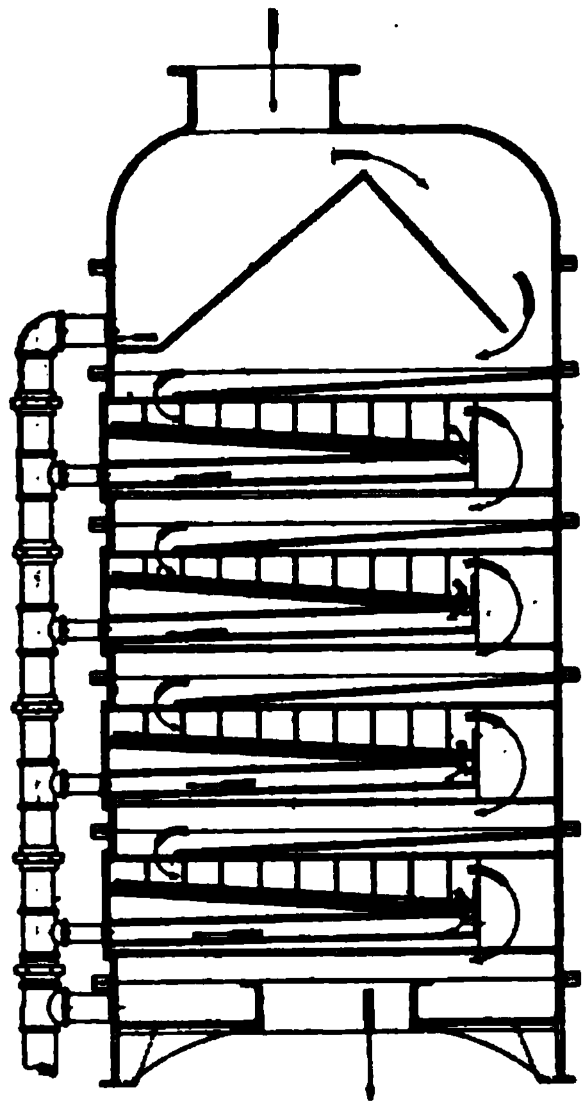


FIG. 318. Loew Grease Extractor.

In a non-condensing plant where the exhaust steam is used for heating purposes the oil separator is ordinarily placed in the main exhaust pipe just before it enters the heating system. Where several branches enter one main it is not customary to place a separator in each branch, one large separator located as above being sufficient.

In condensing plants oil separators are seldom installed except where surface condensers are used, in which case the separator may be placed anywhere between the engine and condenser. In case a vacuum heater is used the separator may be placed on either side of the heater, depending upon the type of separator. If the separator is of the "jacket cooling" or "spray" type, it may be placed between the engine and the vacuum heater; if, however, it is of the "baffle-plate" type, the oil will be more efficiently removed if the separator is placed between the heater and condenser so that it will get the benefit of the moisture formed in the heater. In the latter location, however, the separator will not prevent the oil from fouling the heater tubes.

Where a jet condenser is used and water is taken from the hot well, the hot well itself acts as an oil separator. (Trans. A.S.M.E., 24-1144.)

All separators, steam and oil, should be provided with gauge glasses and should be thoroughly drained and the drainage should be automatic.

307. Exhaust Heads. — The function of the exhaust head is the elimination of oil and water from steam exhaust before permitting it

to be discharged into the atmosphere. Unless removed, the water and oil rot the roofs and walls in summer and pollute the atmosphere surrounding the plant. The exhaust head also acts as a muffler reducing the noise of the escaping steam. Exhaust heads are built on the same principle as steam and oil separators and most separator builders manufacture them. Fig. 319 shows a section through a typical exhaust head. The condensation is ordinarily drained to waste, though with proper purification it may be returned to the boiler. With an efficient oil separator in the exhaust line the condensation in the exhaust head may be returned directly to the boiler without further purification.

Fig. 319. A Typical Exhaust-Head.

Live-steam separators are proportioned so that it is only necessary, in the average installation, to specify the size of pipe, the type of engine, the steam pressure, and the style, whether horizontal or vertical. Gauge glasses, gauge cocks, and companion flanges are usually provided by the maker. In some cases the capacity of the reservoir is also specified. In specifying oil extractors the following additional data are necessary for an intelligent choice: the number of engines and pumps exhausting into the line, the location of the separator, the steam pressure, *velocity*, and the quality and quantity of cylinder oil used. A guarantee of efficiency and of material and workmanship is often demanded.

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Tests of Six Steam Separators: Power, July, 1891, p. 9; Engr. News, Vol. 26, p. 233.

Test of a "Utility" Oil Eliminator: Engr. Rec., May 2, 1903; Engr. Rev., May, 1903.

The Hot Well as an Oil Extractor: Trans. A.S.M.E., 24–1144.

308. Drips. — No matter how thoroughly a steam pipe or reservoir may be covered with insulating material considerable condensation takes place. With the best covering this loss approximates one-sixth of a pound of steam per square foot of pipe surface per hour for steam pressures of one hundred pounds, and runs as high as one pound of steam for bare pipes. See Table 80 for results of experiments on the loss of heat from bare pipes, and Table 81 for data on the efficiency of pipe coverings. In addition to this water of condensation, from ½ to 2 per cent of moisture is carried over by the steam from the boiler. This water, unless thoroughly removed, is a constant source of danger to the engines and causes water hammer and leaky joints in the piping.

A joint on a steam pipe may safely withstand a steam pressure of 100 pounds without leaking and still leak badly under a water pressure of half that amount. This is due to the fact that the steam with its high temperature causes the pipe to expand, thus insuring a tight joint, while the entrained water (which cools as it collects) causes the pipe to contract and allows a leak.

The entrained water and water of condensation are usually spoken of as "drips." Drips may be divided into two classes, low pressure and high pressure.

309. Low-Pressure Drips. — Low-pressure drips include the steam condensed in exhaust steam feed heaters of the closed type, exhaust steam piping, receiver barrels, steam chests, and exhaust heads. As

these drips are impregnated with oil and are useless for boiler feed without purification, they are usually discharged to waste. Most city

FIG. 320. Closed Heater Installation for Abstracting Heat from Oily Drips.

ordinances require the drips to be cooled to 100 degrees F. before being discharged into the sewer. In this case they must be first discharged into a tank and permitted to cool. This tank must be vented to the atmosphere to prevent back pressure. Fig. 320 shows an installation in which the heat asbtracted from the drips, etc., is used to heat the feed water. The drips from the throttle valve and steam chest in a non-condensing plant are ordinarily discharged into the exhaust pipe as shown in Fig. 321. In a condensing plant the throttle drips are piped to a trap or to the free exhaust pipe. The returns from a steam-heating system are sometimes classified as low-pressure drips. They are invariably returned to the boiler.

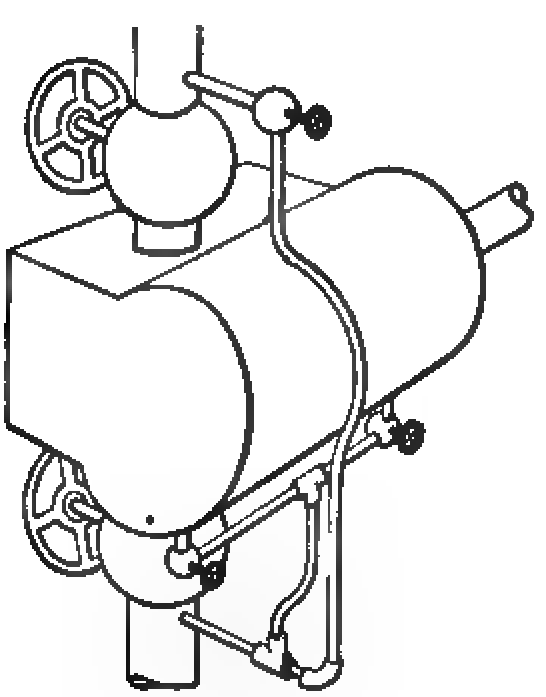


FIG. 321. Simple Method of Draining Drips.

In small plants all the low-pressure drips may be connected to one large pipe and this pipe in turn to a single trap, provided there is but

little difference in pressure in the various drip pipes. In case of different pressures separate leads should be run to waste or traps.

The drips from the receiver and vacuum heater barrels in a condensing plant are oftentimes under less than atmospheric pressure, and sometimes the pressure varies from a slight vacuum to 10 or 20 pounds gauge, and consequently cannot be disposed of as described above. If possible, the heaters and receivers should be placed so as to drain into the condenser (see Fig. 334). Should this arrangement prove impracticable, the barrels may be drained by a trap especially arranged as shown in Fig. 335.

310. Size of Pipe for Low-Pressure Drips. — In the average exhaust-steam feed-water heater one pound of steam in condensing gives up approximately 1000 heat units. This will heat about 6 pounds of water from 60 to 200 degrees F. Hence the area of the drip which carries the water of condensation from the closed heater need be but $\frac{1}{6}$ that of the feed pipe. In no case, however, should a pipe smaller than $\frac{1}{2}$ inch in diameter be used. Should the same pipe be used for both exhaust head and heater drips, an area of $\frac{1}{4}$ area of feed pipe would prove of ample capacity. In practice it is customary to use the size of pipe conforming with the outlet furnished by the manufacturer of the apparatus, and only when several pieces of apparatus are connected to one main are calculations made for the size of this main.

The drip pipe from the throttle valve is ordinarily $\frac{1}{2}$ inch diameter irrespective of the size of steam pipe; this is also true of the steam-chest drip.

311. High-Pressure Drips. — High-pressure drips consist of those which are condensed under practically boiler pressure and include the steam condensed in steam pipes, cylinder jackets of engines, reheating coils of receivers, and separators. Being free from oil and containing considerable heat, they are usually returned to the boiler. Drips may be returned to the boiler automatically by means of

1. Steam traps.
2. Holly steam loop.
3. Pumps.

312. Classification of Steam Traps. — Steam traps may be divided into two classes, depending on their use, — return and non-return. Both of these two classes may be subdivided into five types according to the principle of operation, viz.:

I. Float.	III. Bowl.
II. Bucket.	IV. Expansion.

V. Differential.

Return Traps.

Traps which receive the condensed steam and return it to a boiler having considerably higher pressure than that acting on the returns are known as *return traps*. They are made in a great variety of styles. The general principle of operation is shown in Fig. 331 and described in paragraph 318.

Non-Return Traps.

Non-return traps, as the name implies, are used where the water of condensation is not returned to the boiler but is discharged into any receptacle having less than boiler pressure.

CLASSIFICATION OF A FEW WELL-KNOWN STEAM TRAPS.

- Steam Traps
  - Float
    - McDaniel.
    - Cookson.
  - Bucket
    - Acme.
    - Albany.
  - Dump
    - Bundy.
    - Morehead.
  - Expansion
    - Metal
      - Columbia.
      - Geipel.
    - Volatile-Fluid
      - Dunham.
      - Heintz.
  - Differential
    - Flinn.
    - Siphon.

FIG. 322. McDaniel Float Trap.

313. Float Traps. — Fig. 322 shows a section through a McDaniel improved trap, illustrating the principles of the float type. A hollow sphere C of seamless copper pivoted at E rises and falls with the change of water level in the vessel. The discharge valve M is operated by the

float. When the trap is empty the float is in its lowest position and the discharge valve is closed. Water of condensation flows into the trap by gravity through opening *D* to a certain depth, when the float opens the discharge valve and the steam pressure acting on the surface of the water forces it through outlet *S* to tank or atmosphere. After the water is discharged the float closes the valve and permits the condensation to collect again. A gauge glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is a danger of considerable steam leakage through the discharge valve, due to unequal expansion of valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous, since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present. When the trap is working lightly, this adjustment is apt to throttle the area and create such a high velocity of discharge as to cause a rapid wear of valve and seat. This defect is more or less evident in all steam traps discharging continuously. For this reason all wearing parts should be accessible and readily replaced.

Fig. 323. Acme Bucket Trap.

314. Bucket Traps. — Fig. 323 shows a section through an "Improved Acme" steam trap. The water of condensation enters the cast-iron vessel at *A*, filling the space *D* between the bucket *E* and the walls of

the trap. This causes the bucket to float and forces valve *V* against its seat (valve *V* and its stem being fastened to the bucket as indicated). When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing valve *V* from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water through the annular space *H* to discharge opening *G*. When the bucket is emptied it rises and closes valve *V* and another cycle begins. By closing valve *R* the trap is by-passed and the condensation blows directly through passage *C* to discharge *G*. The discharge from this type of trap is intermittent.

315. Dump or Bowl Traps. — Fig. 324 shows sections through a Bundy bowl trap of the " return " type. The water enters the bowl

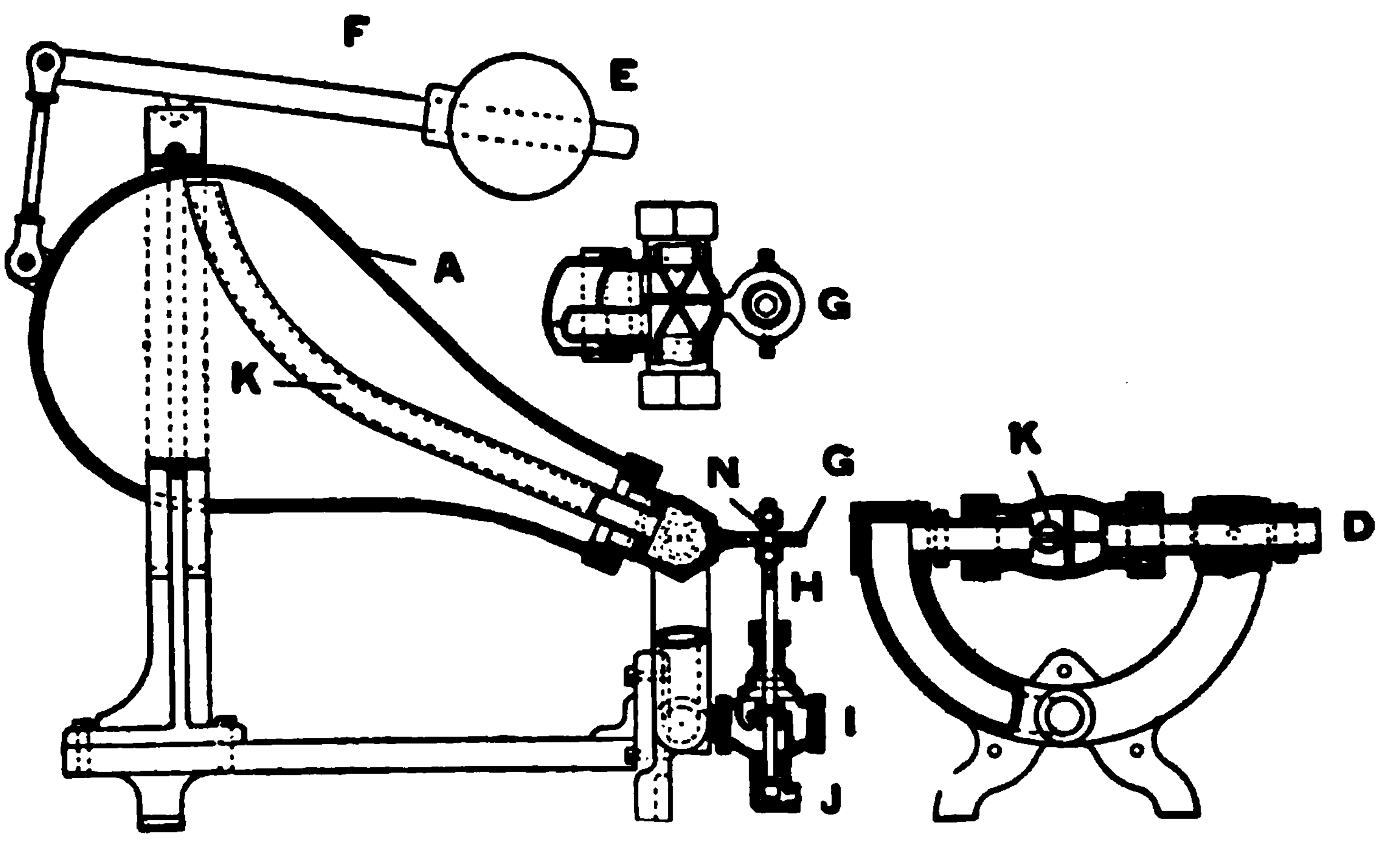


Fig. 324. A Typical Tilting Trap.

through trunnion *D* and rises until its weight overbalances counterweight *E* and the bowl sinks to the bottom. As the bowl sinks, arm *G*, which is a part of the bowl, rises and engages the nuts *N* on valve stem *H* and opens valve *I*, thus admitting live steam pressure on to the surface of the water. The trap then discharges like all others. After the water is discharged weight *E* sinks and raises bowl *A*, which in turn closes valve *I*, and the cycle begins again. Bowl traps are necessarily intermittent in their discharge.

Fig. 335 shows the application of a bowl trap to a receiver where the drips are under a vacuum, and Fig. 336 a similar application to an engine receiver where the pressure varies from less than atmospheric pressure to a pressure of 40 or 50 pounds.

316. Expansion Traps. — Expansion traps may be divided into two groups:

(1) Those in which the discharge valve is operated by the relative expansion of metals and

(2) Those in which the action of a volatile fluid is utilized.

Expansion traps will never freeze, as they are open when cold and all the water drains out before the freezing temperature is reached.

Since traps of this type have little capacity for holding water, 5 to 10 feet of pipe should be provided between the trap and the pipe to be drained in order that the condensation may collect and cool.

Fig. 325 shows the general appearance of a Columbia expansion trap in which the valve is operated by the expansion of metallic tubes.

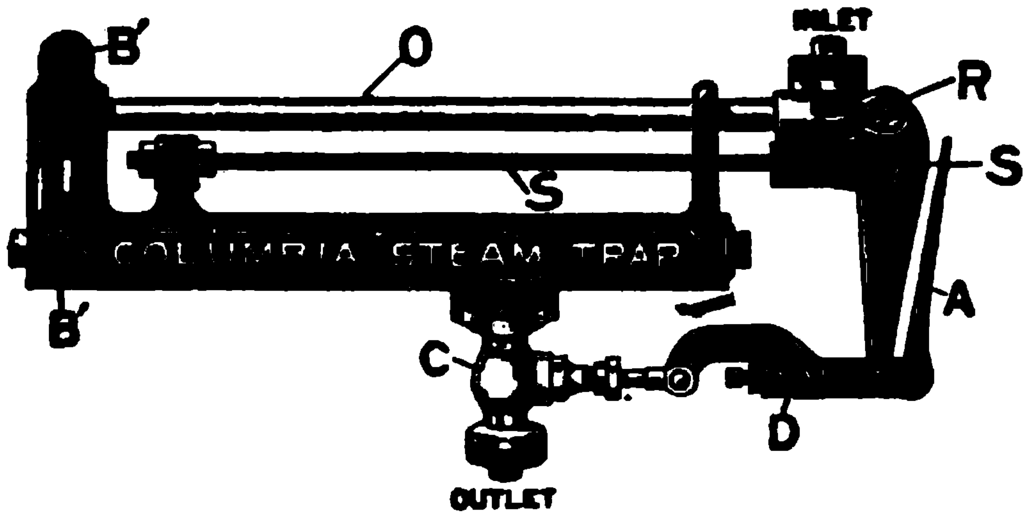


FIG. 325. A Typical Expansion Trap.

Water gravitates to the trap through opening marked "inlet," passes through *brass* pipe *O*, then downward to the main body of the valves and back to outlet valve *C*. Below pipe *O* and parallel to it is an *iron* rod *S*, at the end of which is the support or fulcrum of lever *R*. The lower end of this lever is connected to the stem of the valve *C*, so that any movement of the lever is communicated to it. When the trap is cold, valve *C* is open and all water of condensation passes out. The moment steam enters the pipe *O* it expands. The amount of expansion is multiplied several times by the action of the lever *R*, so that the movement of the valve is much greater than the expansion of the pipe *O*. The compensating spring *D* prevents the brass tube from damaging itself by excessive expansion. Lever *A* permits the trap to be blown through by hand.

Fig. 326 shows a section through a Geipel trap in which the valve is operated directly by the expansion of two metallic tubes and the movement is not multiplied by levers as with the Columbia. The lower or brass pipe constitutes the inlet, and is connected to the vessel to be drained; the upper or iron pipe is the outlet for discharge. The two pipes form the sides of an isosceles triangle, the base *F* of which is

rigid, while the apex A is free to move in a direction at right angles to the linear expansion of the tubes. When cold, the brass pipe is contracted and the apex, in which the valve seat is placed, is moved down so that the valve is open and the water is discharged. As soon as steam

FIG. 326. Geipel Expansion Trap.

enters the brass pipe the latter expands and forces the valve seat against the valve. The trap may be adjusted for any pressure by means of the lock nuts E. When it is desired to blow through, the valve may be operated by hand by pressing the lever.

Fig. 327 shows a section through a Dunham trap. It operates upon the expansion principle, utilizing a fluid of a volatile character as its

FIG. 327. Dunham Expansion Trap.

motive force. The corrugated bronze disk B is filled with a volatile fluid, and expands and contracts according to the pressure exerted by the fluid. The water enters at the top, surrounds disk B and passes through valve opening D to discharge outlet at E. As soon as steam strikes the disk B the volatile fluid flashes into a vapor and causes the disk to expand. This expansion forces valve D against its seat and the discharge ceases. The valve will remain closed until the condensation collects and cools the disk B, which then contracts, opens

the valve, and condensation enters as before. The adjustment, however, is such that the discharge may be made continuous instead of intermittent.

The Dunham trap is claimed to be the smallest trap of its capacity on the market. The 1-inch size, having a capacity for draining 10,000 lineal feet of 1-inch pipe under 60 pounds pressure, weighs but 5 pounds and may be connected to the pipe line as if it were a globe valve.

Fig. 328 shows an internal view of a Heintz steam trap. This works on the principle of the volatile fluid expansion trap but in

Fig. 328. Heintz Expansion Trap.

a different manner from any of those described above. The requisite movement is obtained by the elongation and contraction of the extremities of a bent metallic tube *T* filled with a highly volatile fluid. This tube is inclosed in a cast-iron box and presses against the point of regulating screw *P*. The other extremity of the tube carries the valve and is free to move under the action of the variations of temperature. Spring *S* has no connection with the action of the trap. It is used as a simple means of holding one end of the expansion tube on its pivot. The trap operates as follows: Water enters at *I*, surrounds the tube *T* and passes through the valve to the discharge outlet *O*. As soon as steam enters the chamber the volatile fluid in the tube flashes into a vapor and the pressure thus created tends to straighten out the tube; this forces the valve against its seat and the discharge ceases. As the trap cools the tube returns to its normal position and the discharge valve is opened, thus permitting the condensation to drain out. The adjustment permits of continuous or intermittent discharge and of variable pressures.

317. Differential Traps. — Fig. 329 shows a cross section through a Flinn differential trap. The column of water *X* acting on diaphragm *D* closes valve *V*. The water entering pipe *E* and the action of the spring equalize column *X* and open the valve.

Describing the action in further detail, the water of condensation enters at *A*, fills lower chamber *Y*, pipe *X*, and receiving chamber *C* up to the level of the top of pipe *E*. This column of water acting on the under side of the diaphragm *D* forces the valve to its seat against the counter pressure of the spring *S*. Any additional water that enters the trap overflows through pipe *E*, filling chamber *F* and pipe *E* to a point about midway of its height, where the effect of the column of water in pipe *X* is balanced. The pressure on each side of the diaphragm is then equal, the short column in pipe *E*, aided by the spring, balancing the pressure of the longer column in pipe *X*. Any further increase in the height of the water in pipe *E* causes a depression of the valve *V*, which allows water to escape until the column has fallen to a level a little below the middle of pipe *E*, when this valve closes again. This action is repeated at intervals according to the quantity of water entering the trap. So long as the water keeps coming in sufficiently large quantities the valve remains wide open.

FIG. 329. Flinn Differential Trap.

Fig. 330 gives a general view of a siphon trap which is much used in draining low-pressure systems, as, for example, the separator in an exhaust steam heating system. It consists essentially of two legs *A* and *B*, which may be close together or any distance apart but the lengths of which must be sufficiently great to prevent pressure acting through pipe *I* from forcing the water out of *B*. *C* is a vent pipe extending to the air to prevent siphoning; *O* is the discharge for the condensed steam. In ordinary operation the leg *B* is filled with water which is constantly overflowing, and *A* with steam and water, the

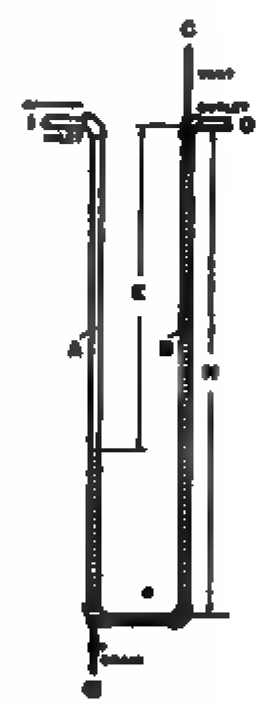


FIG. 330. Simple Siphon Trap.

total pressure in both legs being equal. The siphon trap is applicable for low pressure only, as it requires approximately 2.3 feet of vertical space E for each pound per square inch pressure in the pipe. The maximum allowable head is represented by vertical distance N.

318. Location of Traps. — Wherever possible a trap should be located so that the condensation will flow into it by gravity. This will insure positive drainage. Sometimes, however, the coils, cylinders, or pipes to be drained are located in a pit or trench or lie on a basement floor where it is impossible to set the trap so as to receive the drains by gravity without placing it in an inaccessible position. With very low pressures this is often unavoidable, but with pressures of five pounds or more the trap may be placed above the point to be drained. If a trap is set in an exposed place a drain should be provided at the lowest point to free the pipe of water when steam is shut off. A dirt catcher or strainer should be placed in the pipe leading to the trap to prevent scale, etc., from reaching the valve. All pockets and dead ends should be drained, and no condensation should be allowed to accumulate. High and low-pressure drips should be kept separate. All tanks should have gauge glasses.

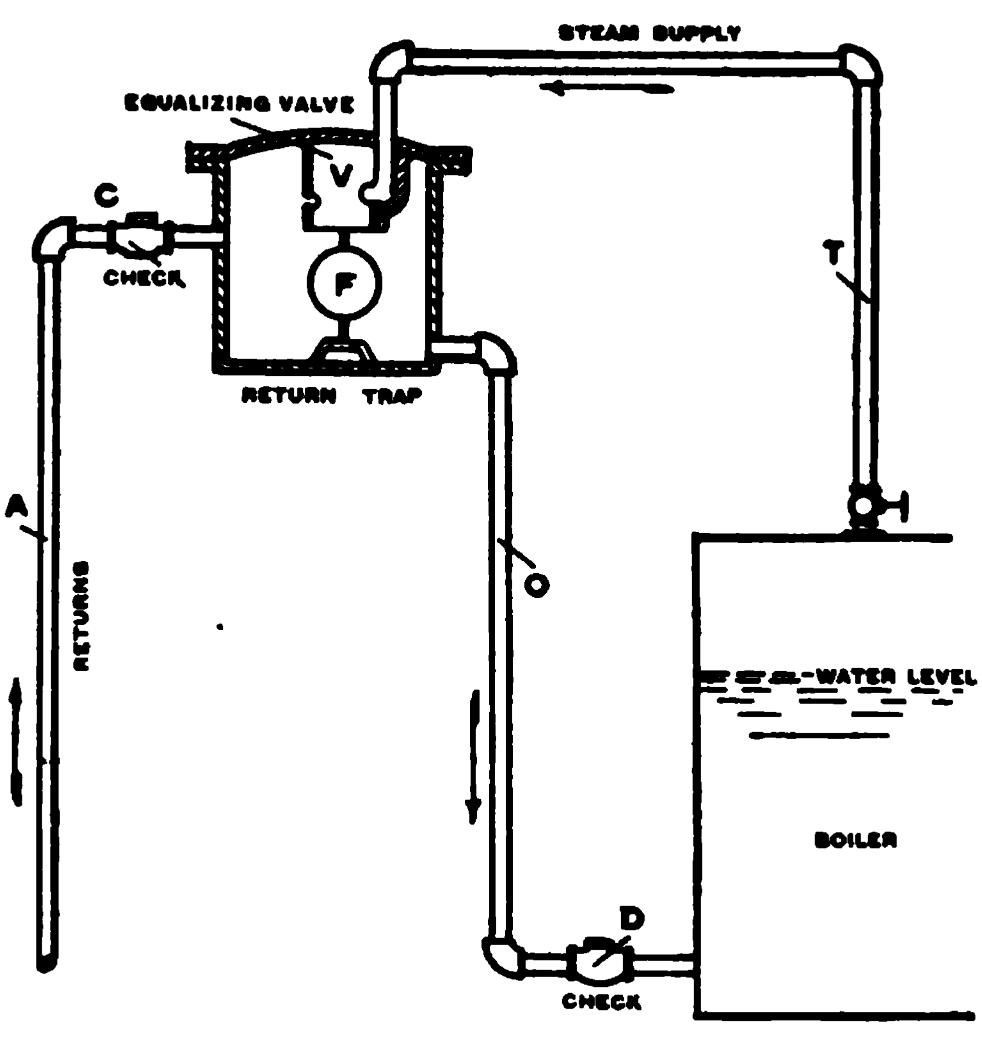


Fig. 331. Return Trap.

Fig. 331 shows the application of a float trap for automatically returning water to the boiler. For this purpose the trap must be placed three feet or more above the water line in the boiler, so that the water may gravitate to it. Water is forced into the trap from the returns through pipe A until it reaches a level where the float opens the equalizing valve V and permits steam from the boiler to enter the trap, thus equalizing the pressures. The water then flows into the boiler by gravity through check valve D. At the end of discharge the float closes the equalizing valve and another cycle begins. Check valve C prevents the water from being forced back to the return pipe. If the pressure in the return pipe A is not sufficient to force the water into

the trap, a pump or another trap may be used to effect this result. Practically any high-pressure trap may be converted into a return trap by proper installation and an "equalizing" valve.

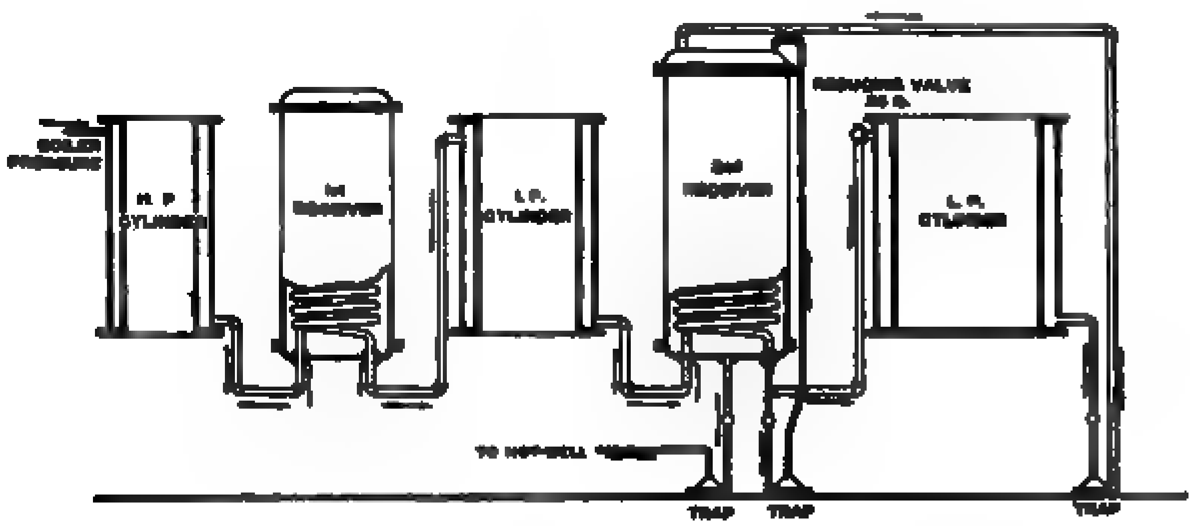


FIG. 332. Drainage System for Jackets and Receivers of Triple Expansion Pumping Engines.

Figs. 332 and 333 show different applications of steam traps to the receiver coils and jackets of triple expansion pumping engines. The drawings are self-explicit.

FIG. 333. Drainage System for Jackets and Receivers of Triple Expansion Pumping Engines.

319. Drips under Vacuum. — Conditions frequently make it necessary to remove condensation from apparatus working under a vacuum, as, for example, a primary heater.

The simplest method is to pipe the drips to the condenser and permit the condensation to gravitate to it as in Fig. 334. Where this is impracticable, as in an installation with the condenser above the

heater, a steam trap is usually employed. Fig. 335 shows the application of a Bundy trap to a vacuum or primary heater. A close-fitting weighted check valve *W*, set to open outwards prevents intake of air through the discharge pipe while the trap is filling. Connection *E* is made from the vent underneath the valve stem *V* back to the heater so as to equalize the pressures. The operation is as follows: Condensation gravitates from the heater through check *C* to the body of the trap, the check *W* being closed. When the bowl is full enough to overcome the weight of the counterbalance, it sinks and opens up the live-steam valve *V*. This admits steam to the trap through pipe *D*, which in

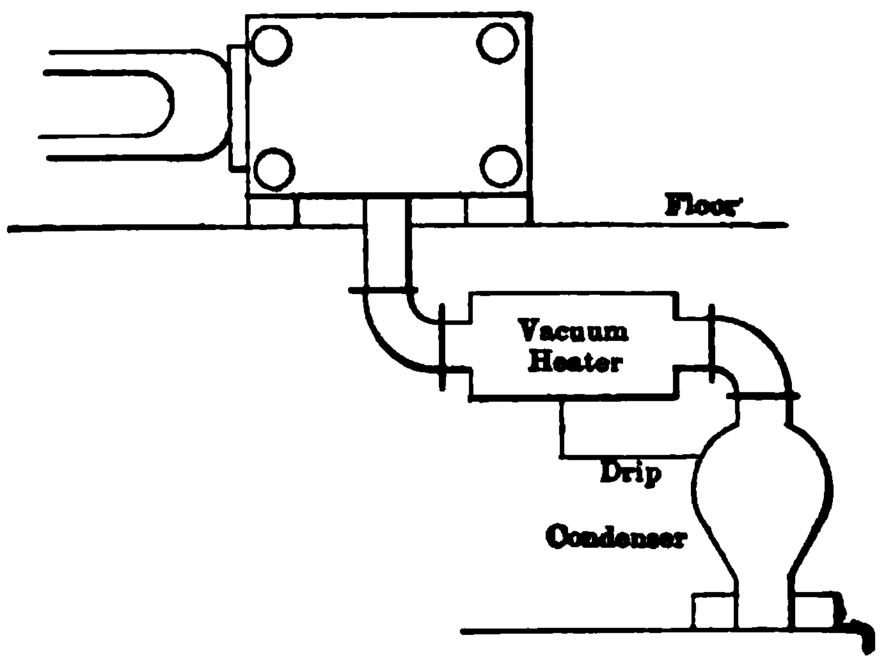


FIG. 334. Gravity Drainage; Vacuum Heater.

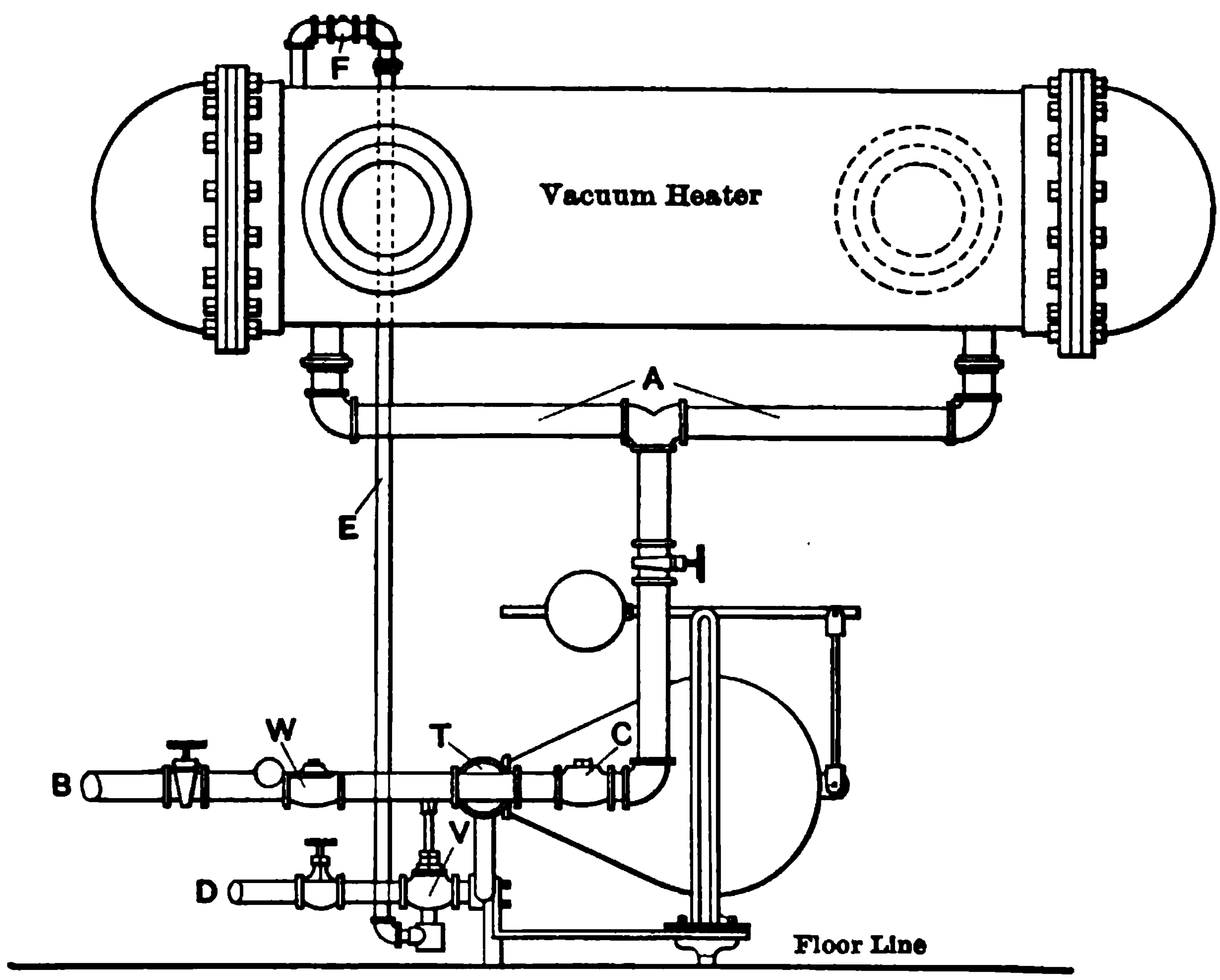


FIG. 335. Method of Draining Heater under Vacuum.

turn closes check *C* and forces the water past the weighted check *W* to the discharge tank. After the water is discharged the bowl returns

to its original position and closes valve V, the weight closes check W, the vent check equalizes the pressure in the bowl and heater, and condensation gravitates to the trap again.

320. Drips under Alternate Pressure and Vacuum. — Occasionally the load on an engine is of such a character that the pressure in the receiver alternates from a pressure of 30 or 40 pounds absolute to a vacuum of varying degree. Where the periods of vacuum operation are very few and of short duration, as in the average installation, no attention is paid to the vacuum and the condensation is removed by a trap in the ordinary way. If, however, the periods are of sufficient duration and frequency, the ordinary method is not applicable and the arrangement shown in Fig. 336 may be used. The trap is placed

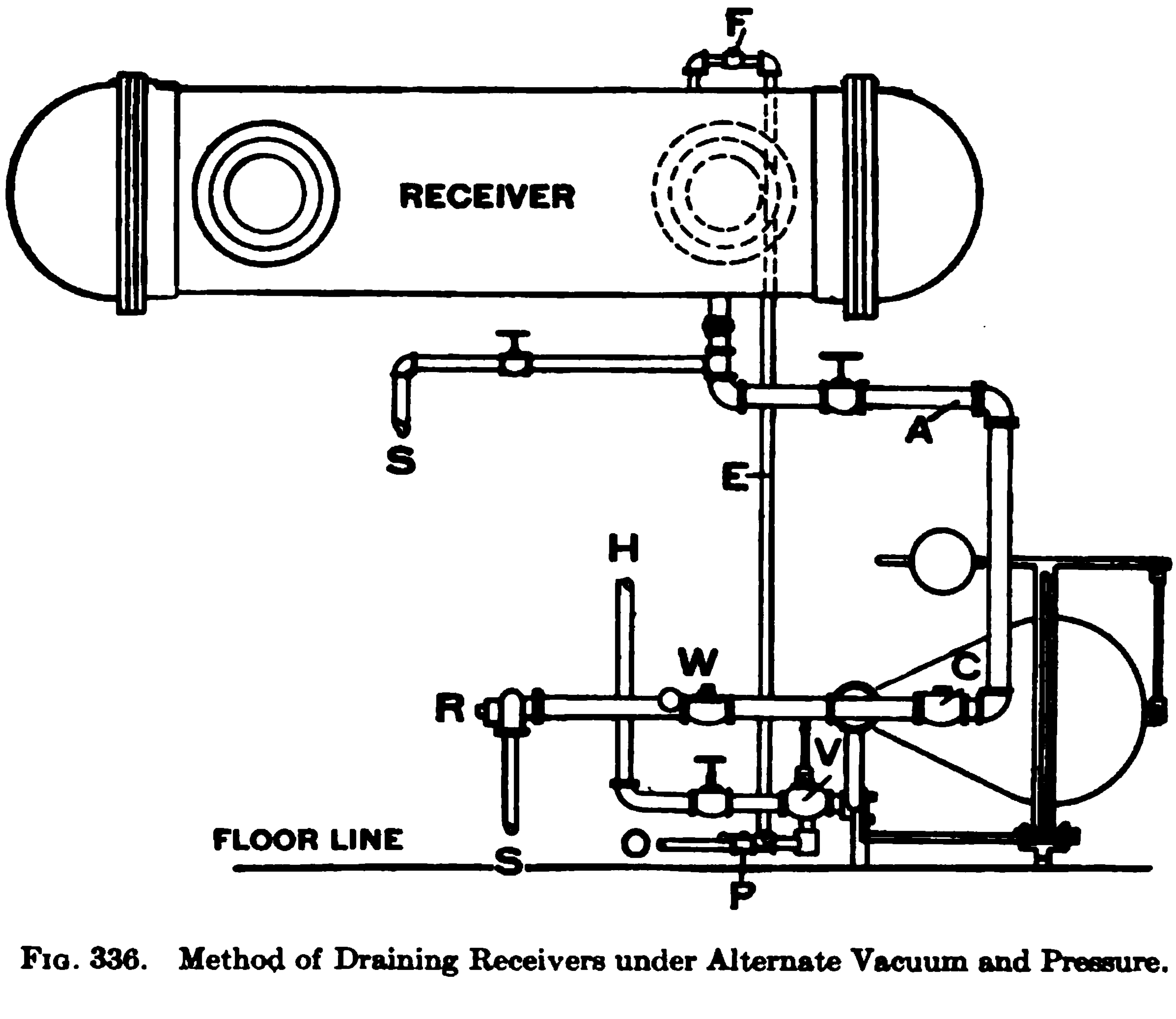


FIG. 336. Method of Draining Receivers under Alternate Vacuum and Pressure.

below the receiver as indicated. The delivery pipe is provided with a weighted check or resistance valve W set so as to open outwards from the trap, also a spring water relief valve R. Another weighted check P is placed in the line leading from the vent to the atmosphere, and a plain check C in the line leading back into the receiver. This arrangement of valves permits the venting of the trap after discharge and effectually excludes air from the trap when there is less than atmospheric pressure on the receiver. With the relief valve set to open at a

pressure in excess of the maximum receiver pressure it acts as a "stop" in the pipe and the water must enter the trap. When the trap discharges, the live steam supplied through the pipe attached to the steam valve forces the water through the weighted check and relief valves into the sewer or receiving tank. When working with a vacuum, the pressures in receiver and trap are equalized through the vent connection and the condensation flows into the trap by gravity. The operation of discharge is the same as in the case of pressure.

321. The Steam Loop. — Fig. 337 illustrates the principles of the "steam loop" for automatically returning high-pressure drips to the

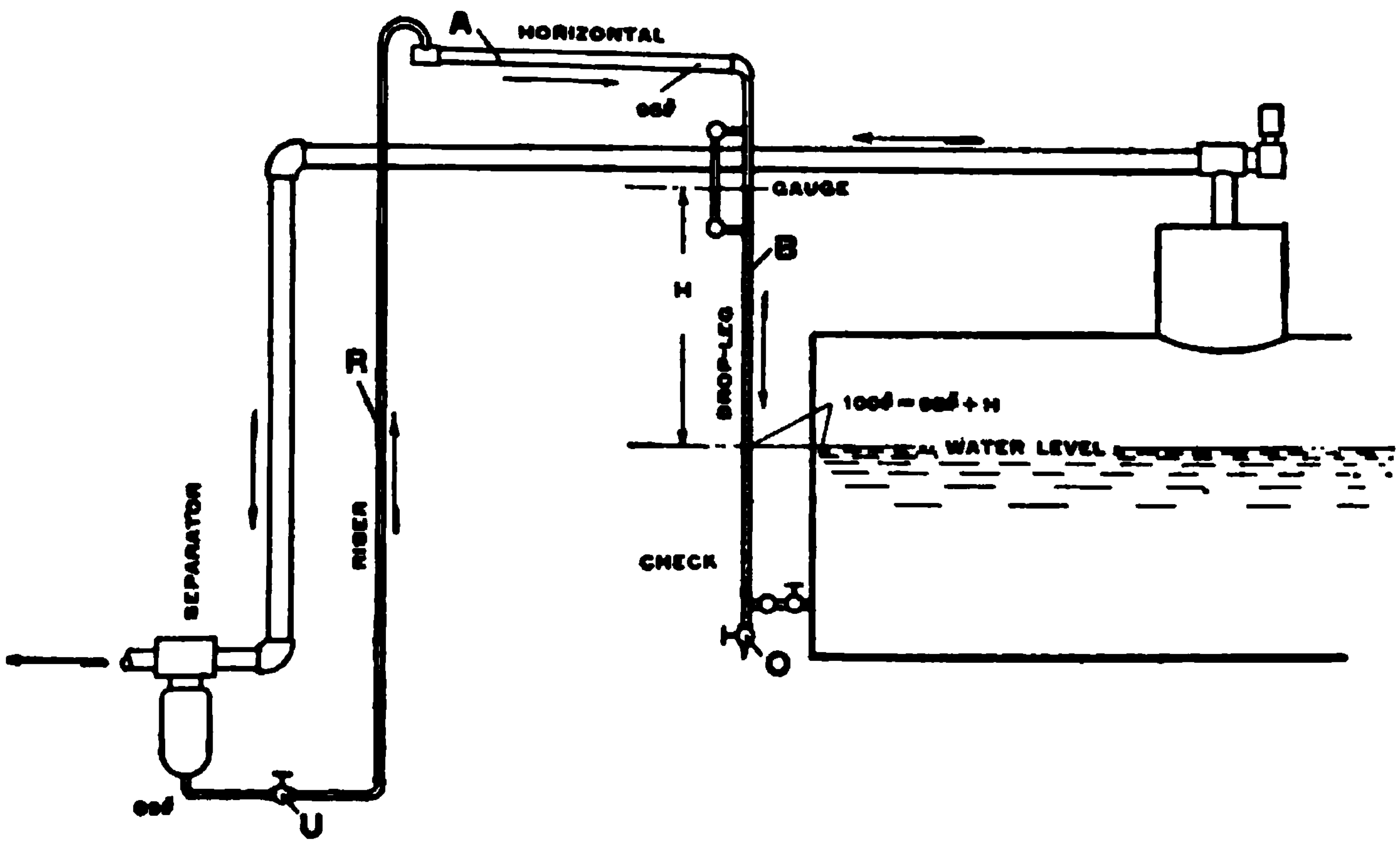


FIG. 337. General Arrangement of the Simple "Steam Loop."

boiler. In the figure the loop is returning the condensation from a steam separator to a boiler above the level of the separator. The apparatus is very simple, consisting of a horizontal and two vertical lengths of plain pipe placed as indicated. Pipes R and B may be covered but "horizontal" A is left uncovered, as its function is that of a condenser. The operation is as follows: Circulation is first started by opening stop valve O at the bottom of the drop leg until steam escapes. The valve is then closed and the steam in the horizontal A condenses and gravitates to the drop leg B. On account of the slight reduction in pressure in the horizontal a mixture of spray and steam flows from the separator chamber to the horizontal, and, condensing, gravitates to the drop leg. The column of water in the drop leg rises until its static head balances the difference in pressure in the riser R and the horizontal. In other words, a decrease in pressure in the horizontal produces similar effects on the contents of the riser and drop leg but in a degree inversely proportional to their densities.

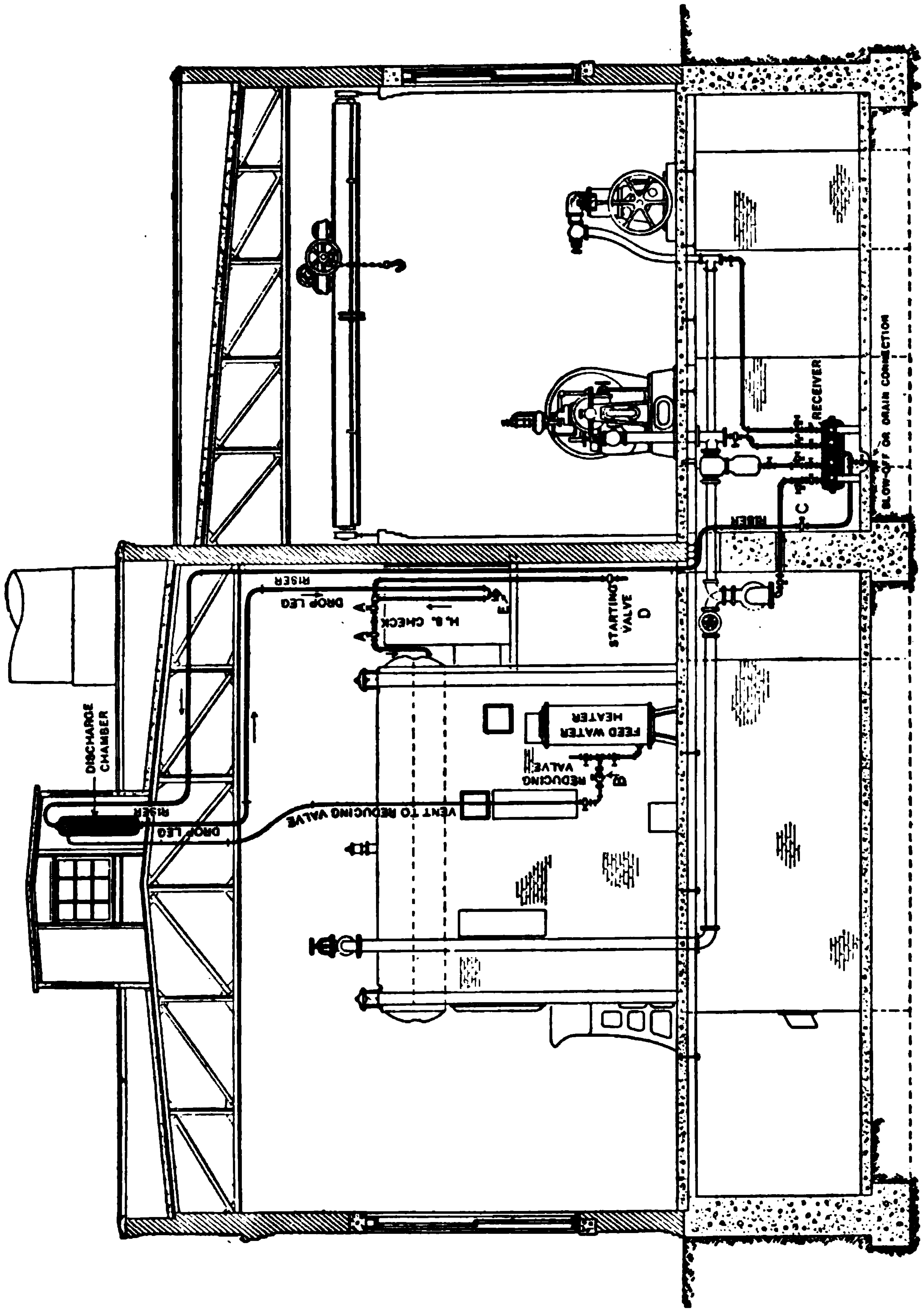


Fig. 338. General Arrangement of the Holly Loop.

Any further accumulation causes an equal amount to pass from the bottom of the column to the boiler, since the pressure in the boiler is then less than that at the bottom of the column, that is, the steam pressure on the top of the water column plus the hydrostatic head H is greater than the pressure in the boiler. Once started the process is continuous and requires no further attention.

322. The Holly Loop. — In the application of the steam loop where many points requiring drainage are connected to many boilers and conditions are more complex, some method other than the simple one of radiation may be advisable to secure the necessary lower pressure at the top of the loop. Such a method is illustrated in Fig. 338. This arrangement differs from the simple loop in that all condensation first gravitates to a "Holly" receiver (shown in detail in Fig. 339) before passing into the "riser." The receiver is placed below the lowest point to be drained and serves as a storage for large or unusual quantities of water and enables the riser to act at a constant rate independent of variable discharge into the receiver.

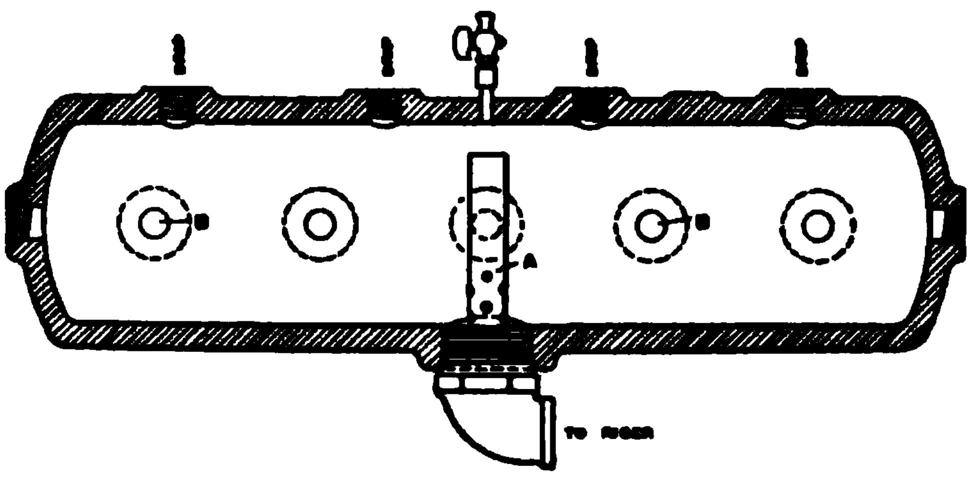


Fig. 339. Holly Receiver.

Furthermore, the lower pressure in the discharge chamber necessary to secure the lifting of the mingled steam and water through the riser, instead of being created by condensation as in the simple loop, is produced by a reducing valve B discharging into the feed-water heater. The operation of the Holly loop is as follows: Circulation is started by opening valve D until steam appears. Valve D is then closed and the reducing valve is put into commission. Condensation from separators, traps, and pipes gravitates to the "receiver," from which it is forced into the "riser" in the form of a spray. The spraying effect is produced by a series of holes drilled in pipe A, Fig. 339. From this receiver the spray and moisture rise to the "discharge chamber," on account of the lower pressure at that point, where the steam and entrained water are separated, the water gravitating to the bottom of the chamber and thence to the drop leg, and the steam discharging through the reducing valve into the heater. The principles of operation are exactly the same as in the simple steam loop.

323. Returns Tank and Pump. — Low-pressure drips in connection with heating systems may be returned to the boiler along with the condensation from the heating system by a combined pump and receiver

as shown in Fig. 340. The height of water in the tank controls the operation of the pump through the medium of a float and throttle valve. This combination of float and balanced throttle valve is sometimes called a "pump governor." In the illustration the pump forces the returns through a closed heater before delivering them to the boiler,

FIG. 340. Returns Tank and Pump.

though they are oftentimes returned directly. The tank is vented to the atmosphere to prevent it from becoming "air bound." The cold-water supply or make-up water is sometimes discharged into the receiving tank as indicated. With open heaters the cold supply is ordinarily controlled by a float within the heater itself.

324. Office Building Drains. — In the power plants of tall office buildings the public sewers are often above the basement level, and it is necessary to remove all liquid wastes mechanically.

The Shone pneumatic ejector has been found to serve this purpose effectually. This apparatus is placed in a pit in the basement floor into which all sewage, drips from engines, washings from boilers, and ground water gravitate, and are automatically discharged into the street sewer by means of compressed air.

Fig. 341 gives a sectional view of a Shone ejector of ordinary construction. It consists essentially of a closed vessel furnished with inlet and discharge connections fitted with check valves, *A* and *B*, opening in opposite directions with regard to the ejector. Two cast-iron bells, *C* and *D*, are linked to each other, in reverse positions, the

FIG. 341. Shone Ejector.

rising and falling of which control the supply of compressed air through the agency of automatic valve *E*.

The bells are shown in their lowest position, the supply of compressed air is cut off from the ejector, and the inside of the vessel is open to the atmosphere. The sewage gravitating into the ejector raises the bell *C*, which in turn actuates the automatic valve *E*, thereby closing the connection between the inside of the ejector and the atmosphere and opening the connection with the compressed air. The air pressure expels the contents through the bell-mouthed opening at the bottom and the discharge valve *B*, into the main sewer. Discharge continues until the level falls to such a point that the weight of the sewage retained in the bell *D* is sufficient to pull it down, thereby reversing the automatic valve. This cuts off the supply of compressed air and reduces the pressure to that of the atmosphere.

The positions of the bells are so adjusted that compressed air is not admitted until the ejector is full, and is not allowed to exhaust until emptied down to the discharge level; thus the ejector discharges a fixed quantity each time it operates.

Two ejectors, each of a capacity suitable for handling the average flow of tributary sewage and so arranged that they can work either independently or together, are usually installed at each ejector station.

The main sanitary sewer of the building usually discharges directly into the ejectors, the surface water, drips, etc., being collected in a neighboring sump. The latter is connected to the sanitary sewer through a trap or back-water valve.

CHAPTER XV.

PIPING AND PIPE FITTINGS.

325. General. — The advent of high pressures and superheat is responsible for the elimination of many of the older systems of piping, the tendency being towards greater uniformity in design, particularly in electric central station work. In isolated stations the conditions of operation and installation are so variable that each case presents an entirely different problem. In any system of piping the fundamental object is to conduct the fluid in the safest and most economical manner.

The material should be the best obtainable and the system so flexible that a break-down in one element will not necessitate the closing down of the entire plant. On the other hand, flexibility increases the number of parts and, unless first cost is of little importance, tends to weaken the system as a whole. It is a safe general proposition to say that the best pipe and fittings, irrespective of first cost, will prove the most economical in the end, but few owners of power plants are willing to take this view.

326. Drawings. — An assembly drawing of the entire installation giving the location of all valves and fittings is necessary in order to avoid interference, and particularly where a number of fittings are to be close together. Detailed drawings should also be provided of each division of the piping to facilitate installation, as, for example, the high-pressure steam, the exhaust steam, the feed water, the condensing water, the oil, the heating and the sanitary piping. As a rule, lower and more uniform bids will be obtained from an isometric or perspective sketch, as in Fig. 254, than from conventional plan and elevation drawings, due, no doubt, to the greater ease with which the drawing is interpreted. A complete set of specifications for a piping system is given in paragraph (415), and illustrates the usual practice along this line.

327. Materials for Pipes and Fittings. — The following materials are used in the construction of pipes for steam, water, and gases.

	Average Bursting Tension.
Low-carbon or mild steel	65,000 lbs. per sq. in.
Wrought iron	50,000 lbs. per sq. in.
Cast iron, high grade	20,000 lbs. per sq. in.
Cast steel	50,000 lbs. per sq. in.
Wrought copper	33,000 lbs. per sq. in.
Brass	18,000 lbs. per sq. in.
Special alloys and compounds	15,000–60,000 lbs. per sq. in.

Mild Steel. — The greater portion of the piping in the average steam power plant is of mild steel, lap or butt welded for high pressures and riveted for very low pressures and large diameters. Steel pipe is considerably cheaper than that manufactured from other material, and fulfills practically all requirements for general service.

Wrought Iron. — Pipes manufactured from *puddled* iron, though not as strong in ultimate bursting strength as low-carbon steel pipes, are superior in many ways. Threads are cut more readily and with less power. It is more easily bent without injury and offers more resistance to corrosion. The life of wrought-iron pipe is greater than that of steel pipe under conditions of extreme exposure, or when buried underground, and it is recommended in all cases where corrosion is apt to be severe, as in blow-off pipes, drips, and drains or in pipes not in continuous use. Steam pipes well covered and protected from external moisture are ordinarily of mild steel, as the conditions do not warrant the use of wrought iron, which costs approximately 10 per cent more. Since the term " wrought-iron pipe " is used rather loosely in practice, the manufacturer will generally furnish *mild steel* unless special stress is laid upon the term " wrought iron."

Wrought Iron vs. Steel Pipe: Ir. Age, March 2, 1905, Jan. 18, 1906; Pro. Heat and Vent. Engr., Jan., 1900; Mach., Dec., 1903, p. 191; Am. Mfg. and Ir. World, April 29, 1898; Eng. News, 50–292, 296, 487, 51–62; Eng. Rec., 44–54; Locomotive, Jan. 1, 1906.

Corrosion of Pipes: Trans. A.S.M.E., 18–282; Eng. Rec., 42–194, 43–354, 45–584.

Cast-Iron Pipes. — Cast iron is little used for high-pressure steam, except occasionally in the construction of headers where a number of branches lead into a single pipe, in which case the number of joints is greatly reduced and the cost considerably less than for wrought-iron or steel pipe with numerous fittings and joints. The chief objections to cast iron for high-pressure steam are its weight and lack of homogeneity. It is most used in connection with water service and sanitation.

Cast-Iron Pipe: Jour. New Eng. Waterworks Assn., March, 1907; Engr., Lond., Nov. 7, 1902, p. 454; Power, Jan., 1904, p. 334; Eng. News, 46–216, 48–193, 51–544.

Cast-Steel Pipe. — Cast-steel headers are sometimes used in power plants for highly superheated steam, since the material is not affected by high temperatures to the same extent as cast iron. High first cost and the difficulty of securing castings free from blow-holes have prevented its more general use.

Copper Pipes. — Copper steam pipes were in common use for many years in marine service on account of their flexibility. To increase the bursting strength, pipes above 6 inches in diameter were generally wound

with a close spiral of copper or composition wire. In recent years wrought-iron and steel pipe bends have practically superseded copper for flexible connections. As a rule the use of copper pipes should be avoided on account of the rapid deterioration of the metal under high temperatures and stress variations. The cost is prohibitive for most purposes and this alone prevents it from being seriously considered in the manufacture of pipe.

Copper Pipes: Engr., Lond., April 15, 1898, p. 360, Aug. 11, 1893; Engng., April, 1898; Eng. Rec., July 30, 1898.

Brass Pipes. — Brass is little used in the construction of pipes on account of its high cost. It withstands corrosive action much better than iron or steel and is often used in connecting the feed main with the boiler drum. Special alloys, nickel steel, "ferrosteel," malleable iron, and the like have been used in the manufacture of pipes, and possess points of superiority over wrought iron and steel for some purposes, as for highly superheated steam, but the cost is prohibitive for average steam power plant practice.

Materials for Fittings. — Elbows, tees, flanges, and similar fittings are usually made of cast iron, malleable iron, or pressed steel, though cast steel, "ferrosteel," and other steel compounds are used to a limited extent. Standard cast-iron fittings are recommended for ordinary pressures of 100 pounds per square inch or less, and extra heavy cast-iron fittings for higher pressures. Malleable iron fittings are lighter and neater than cast iron and are extensively used for small sizes of steam and gas pipe.

Manufacture of Pipe: Sci. Am., Dec. 12, 1903; Mach., Feb., 1904, Dec., 1903, p. 191; Eng. News, 50–232, 296.

328. Size and Strength of Commercial Pipe. — Wrought-iron and mild-steel pipe are marketed in standard sizes. Those most commonly used in steam power plants are designated as

(1) Merchant or standard pipe.
(2) Full weight pipe.
(3) Large O. D. pipe.
(4) Extra heavy.
(5) Double extra heavy.

Table 76 gives the dimensions of standard "full-weight" pipe, which is specified by the *nominal* inside diameter up to and including 12 inches and based on the Briggs standard. Pipes larger than 12 inches are designated by the *actual* outside diameter (O. D.), and are made in various weights as determined by the thickness of metal speci-

fied. Manufacturers specify that "full weight" pipe may have a variation of 5 per cent above or 5 per cent below the nominal or table weights, but merchant pipe, which is the standard pipe of commerce, such as manufacturers and jobbers usually carry in stock, is almost invariably under the nominal weight. It varies somewhat among the different mills, but usually lies between 5 and 10 per cent under the table weight. The smaller sizes of merchant pipe, ⅛ inch to 3 inches, are butt-welded and the larger sizes are lap-welded.

Extra heavy and double extra heavy pipe have the same external diameter as the standard, but are of greater thickness and hence the internal diameter is smaller. Taking the thickness of the standard pipe as 1, that of the extra heavy is approximately 1.4 and of the double extra heavy 2.8.

Wrought-iron and steel pipes are ordinarily designed with factors of safety of from 6 to 15, with an average not far from 10. The standard hydrostatic tests to which the various pipes are subjected at the mills are as follows:

	Hydrostatic Pressure, Lbs. per Sq. In.
Standard, butt-welded, ⅛–3 in.	600 to 1,000
Standard, lap-welded, 3–12 in.	500 to 1,000
Extra heavy, butt-welded, ⅛–3 in.	600 to 1,500
Extra heavy, lap-welded, 1¼–12 in.	600 to 1,500
Double extra heavy, butt-welded, ⅛–2½ in	600 to 1,500
Double extra heavy, lap-welded, 1¼–8 in.	1,200 to 1,500

The pressure necessary to burst piping is far above anything likely to occur in ordinary practice on account of the thickness of material necessary to permit of threading, thus:

	Actual Bursting Pressure, Lbs. per Sq. In.
2-inch "standard" mild-steel pipe*	5,800
2-inch " wrought-iron pipe	4,106
10-inch " mild-steel pipe†	3,000
10-inch " wrought-iron pipe †	1,900
10-inch "extra heavy" wrought-iron pipe †	2,700

* *Machinery*, December, 1903, p. 192. † Crane Company, Published Tests.

Riveted Pipes. — For low pressures and large diameters, pipes are constructed of thin sheets of boiler steel with riveted joints, the seams being either longitudinal and circumferential, or spiral. Such pipes are not necessarily limited to large sizes and low pressures, though this is the usual practice.

Pipe fittings are classed as screwed or flanged.

329. Screwed Fittings, Pipe Threads. — For screw connections the ends of pipes and fittings are threaded to conform to the Briggs or United States standard system, as shown in Fig. 342. The end of the

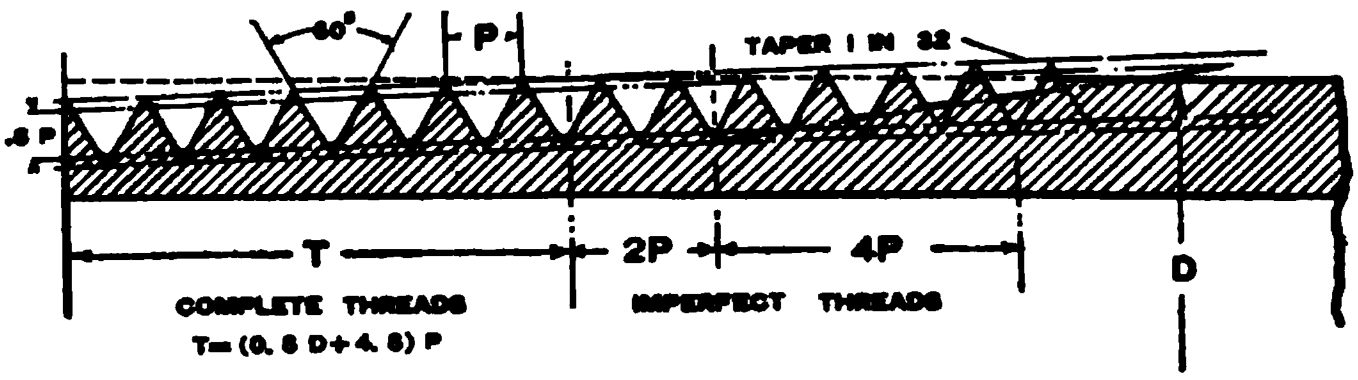


Fig. 342. Standard U. S. Pipe Thread.

pipe is tapered 1 to 32 with the axis, the angle of the thread being 60 degrees and slightly rounded at top and bottom. The proper length of perfect threads is given by the formula

$$T = \frac{(0.8\,D + 4.8)}{n}, \tag{146}$$

in which

T = length in inches.
D = actual external diameter of the tube, inches.
n = number of threads per inch.

The imperfect portion of the thread is simply incidental to the process of cutting. The object of the taper is to facilitate "taking hold" in making up the joint. Table 76 gives the number of threads per inch for various sizes of standard pipe. When properly constructed a screwed joint will hold against any pressure consistent with the strength of the pipe. For example, the ultimate bursting strength of a "standard" 2-inch pipe is about 6000 pounds per square inch, while the stripping strength of the joint (with perfect threads) is 225,000 pounds. The threads, however, are often poorly cut and the parts screwed together improperly cleaned and lubricated, thus causing leakage between the threads.

330. Flanged Fittings. — In cast-iron pipes, valves, tees, and other fittings the flange is always a part of the casting, but for joining the two ends of a steel or wrought-iron pipe the flanges may be fastened to the pipe in a number of ways. Fig. 343, *A* to *H*, illustrates methods most commonly used. In *A* to *C* the pipes are screwed into cast-iron or forged-steel flanges and the two faces, with metallic or composition gasket between, are drawn together by bolts. *A* illustrates the most common and inexpensive of flanged joints, which requires no special tools and can be made up at the place of

TABLE 76.

STANDARD DIMENSIONS OF WROUGHT-IRON AND STEEL STEAM, GAS, AND WATER PIPE.

Diameter.			Nominal Thickness.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Length of Pipe Containing One Cubic Foot.	Nominal Weight per Foot.	Number of Threads per Inch of Screw.
Nominal Internal.	Actual External Diameter.	Approximate Internal Diameter.		External.	Internal.	External.	Internal.	Metal.	External Surface.	Internal Surface.			
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Sq. Inch.	Sq. Inch.	Sq. Inch.	Feet.	Feet.	Feet.	Pounds.	
1/8	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513	.241	27
1/4	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.42	18
3/8	.675	.494	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.559	18
1/2	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14
3/4	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270	1.115	14
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.668	11½
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11½
1½	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.678	11½
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11½
2½	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.739	8
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8
3½	4	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8
4½	5	4.508	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.49	8
5	5.563	5.045	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.762	8
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	23.271	8
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.177	8
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.065	8
11	11.75	11	....	36.914	34.558	108.434	95.033	13.401	.325	.347	1.51	45.028	8
12	12.75	12	....	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	48.985	8

erection. It gives satisfactory results for pressures of 100 pounds or less, but for higher pressures leakage is apt to take place between the threads. The flanges are sometimes made with a long thread and a recess which can be calked with soft metal. A similar joint is made

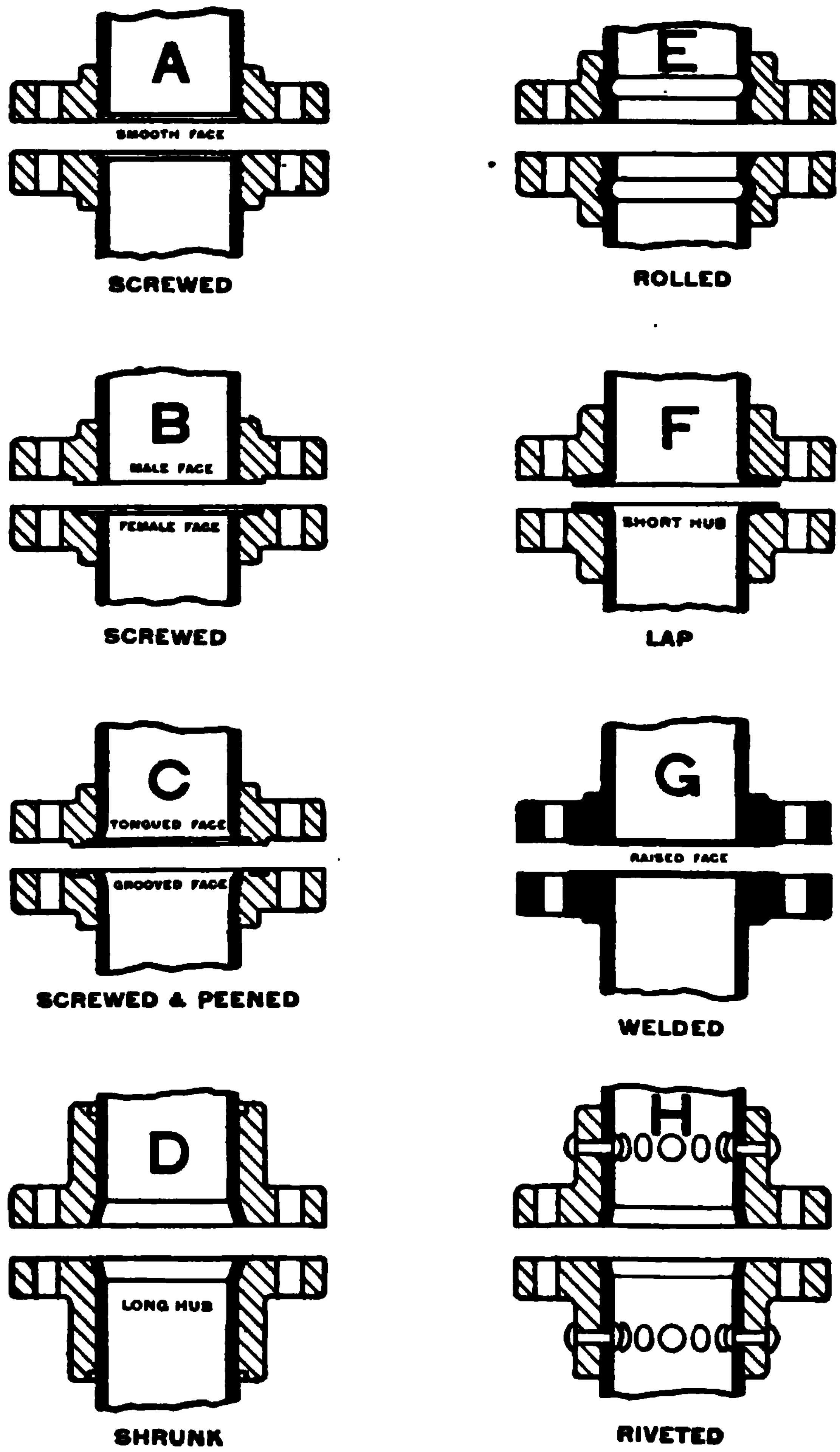


FIG. 343. Types of Pipe Flanges.

with the pipe screwed beyond the face of the flange and the two faced off together, either plane or as shown in *B*, which is known as a *male and female* or *hydraulic* joint. This method forms a very reliable joint, since the ends of the pipe bear on the gasket, and the gasket is pre-

vented from being blown out. An objection lies in the difficulty of opening the line to remove the gasket or replace a fitting. *C* is a modification known as the *tongued and grooved* joint, which uses an extremely narrow gasket. Such flanges may be subjected to severe strains when the bolts are drawn up, owing to the small area of contact. Corrugated copper or steel gaskets are recommended, since soft material is apt to be squeezed out. In *C* the ends of the pipe are *peened,* which is an improvement over the simple screwed joint. *D* illustrates a *shrunk* joint. The flanges are bored for a shrink fit and forced over the pipe when at a red heat. After cooling, the end is beaded over into a recess on the face of the flange and a light cut taken from both. *H* shows a modification in which the hub is riveted to the pipe. *E* illustrates a joint constructed by rolling the pipe into a corrugation in the flange. The end of the pipe is then faced off flush.

One of the best commercial joints is illustrated by *F* and is known as the *lap joint.* The pipe is expanded as indicated and a light cut is then taken from the flared ends to insure a tight joint. The flanges are loose and permit of considerable flexibility in shifting them through various angles. This is sometimes called the *Van Stone* joint.

Pipes with flanges *welded* on the end as in *G* have proved the most reliable of all and though costly are considered the standard for high-pressure and high-temperature work. The faces are ordinarily raised $\frac{1}{32}$ to $\frac{1}{16}$ inch inside the bolt holes and ground to a steam-tight fit, so that thick gaskets are unnecessary.

For moderately high pressures and temperatures any of the joints when well made will prove satisfactory. For extremely high pressures and temperatures the lap or welded joints are preferable.

The comparative costs of various flanges are given in Table 77.

On July 18, 1894, committees of the American Society of Mechanical Engineers, of the National Association of Steam and Hot Water Fitters, and of Manufacturers met and adopted a schedule for dimensions of flanges, known as the "A.S.M.E. Master Steam Fitters'" flange schedule. Flanges dimensioned in accordance with this schedule, Table 78, are applicable for pressures up to 100 pounds per square inch. For higher pressures a schedule shown in Table 79 was adopted June 28, 1901.

Pipe Flanges and Fittings: Power & Engr., Mar. 2, 1909, p. 402, June, 1904, p. 354, Jan., 1904, p. 52, Oct., 1902, p. 36; Am. Mach., April 18, 1901; Jour. Assn. Engr. Soc., Sept., 1904; Jour. Am. Soc. Nav. Engrs., May, 1905; Trans. A.S.M.E., 20–737, 429, 8–29, 347, 14–49, 7–311; Eng. News, 41–183, 323; Eng. Rec., 41–113, 44–209.

Methods of Welding Nozzles, Flanges, etc., to Steam Pipes: Power & Engr., Sept. 28, 1909, p. 518.

TABLE 77.

COMPARATIVE COST OF VARIOUS PIPE FLANGE FITTINGS, 12-INCH PIPE.

(Circular from the Crane Company.)

	Screwed.	Shrunk.	Lap Joint. Long Hub.	Lap Joint. Short Hub.	Lap Joint. No Hub.	Welded.	Rolled.	Single Riveted.
Cast iron	$ 7.40	$16.00	$18.00	......	......	......	$13.00	$21.00
Ferrosteel	8.70	18.40	20.00	......	......	......	16.00	23.40
Malleable iron	9.90	......	......	$22.00	......	......	18.00	......
Cast steel	22.40	28.40	34.00	......	$33.00	......	25.00	33.40
Weldless steel	26.40	32.40	38.00	......	37.00	$41.00	30.00	37.40

Any of the above screwed, shrunk, welded, rolled, or single-riveted flanges can be furnished with male or female face at $1.25 extra.

The screwed or welded flanges can be furnished with tongued or grooved face at $1.25 extra.

Any of the above screwed, shrunk, or single-riveted flanges can be furnished with calking recess at $1.25 extra.

TABLE 78.

SCHEDULE OF STANDARD FLANGES ADOPTED JULY 18, 1894, BY A COMMITTEE OF THE MASTER STEAM AND HOT WATER FITTERS' ASSOCIATION, AMERICAN SOCIETY OF MECHANICAL ENGINEERS, AND VALVE AND FITTING MANUFACTURERS. SUITABLE FOR PRESSURES UNDER 100 POUNDS PER SQUARE INCH.

Size of Flange, Pipe Size × Diam.	Diameter of Bolt Circle.	Number of Bolts.	Size of Bolts, Pressure under 80 Pounds.	Size of Bolts, Pressure 80 Pounds and Over.	Flange Thickness at Hub for Iron Pipe.	Flange Thickness at Edge.	Width of Flange Face.
2 × 6	4¾	4	½ × 2	⅝ × 2	1 in.	⅝	2
2½ × 7	5½	4	½ × 2¼	⅝ × 2¼	1⅛	11/16	2¼
3 × 7½	6	4	½ × 2½	⅝ × 2½	1¼	¾	2¼
3½ × 8½	7	4	½ × 2½	⅝ × 2½	1¼	13/16	2½
4 × 9	7½	4	⅝ × 2¾	¾ × 2¾	1⅜	15/16	2½
4½ × 9¼	7¾	8	⅝ × 3	¾ × 3	1⅜	15/16	2⅝
5 × 10	8½	8	⅝ × 3	¾ × 3	1½	15/16	2½
6 × 11	9½	8	⅝ × 3	¾ × 3	1½	1	2½
7 × 12½	10¾	8	⅝ × 3¼	¾ × 3¼	1½	1 1/16	2¾
8 × 13½	11¾	8	⅝ × 3½	¾ × 3½	1¾	1⅛	2¾
9 × 15	13¼	12	⅝ × 3½	¾ × 3½	1¾	1⅛	3
10 × 16	14¼	12	¾ × 3⅝	⅞ × 3⅝	2	1 3/16	3
12 × 19	17	12	¾ × 3¾	⅞ × 3¾	2	1¼	3½
14 × 21	18¾	12	⅞ × 4¼	1 × 4¼	2	1⅜	3½
15 × 22¼	20	16	⅞ × 4¼	1 × 4¼	2	1⅜	3⅝
16 × 23½	21¼	16	⅞ × 4¼	1 × 4¼	2¼	1 7/16	3¾
18 × 25	22¾	16	1 × 4¾	1⅛ × 4¾	....	1 9/16	3½
20 × 27½	25	20	1 × 5	1⅛ × 5	....	1 11/16	3¾

TABLE 79.

SCHEDULE OF STANDARD FLANGES FOR EXTRA HEAVY STEEL PIPE, FITTINGS, AND VALVES ADOPTED JUNE 28, 1901, BY VALVE AND FITTING MANUFACTURERS. SUITABLE FOR PRESSURES FROM 125 TO 250 POUNDS PER SQUARE INCH.

Size of Pipe, Inches.	Diameter of Flange, Inches.	Thickness of Flange, Inches.	Diameter of Bolt Circle, Inches.	Number of Bolts.	Diameter of Bolts, Inches.
2	$6\frac{1}{2}$	$\frac{7}{8}$	5	4	$\frac{5}{8}$
$2\frac{1}{2}$	$7\frac{1}{2}$	1	$5\frac{7}{8}$	4	$\frac{3}{4}$
3	$8\frac{1}{4}$	$1\frac{1}{8}$	$6\frac{5}{8}$	8	$\frac{5}{8}$
$3\frac{1}{2}$	9	$1\frac{3}{16}$	$7\frac{1}{4}$	8	$\frac{5}{8}$
4	10	$1\frac{1}{4}$	$7\frac{7}{8}$	8	$\frac{3}{4}$
$4\frac{1}{2}$	$10\frac{1}{2}$	$1\frac{5}{16}$	$8\frac{1}{2}$	8	$\frac{3}{4}$
5	11	$1\frac{3}{8}$	$9\frac{1}{4}$	8	$\frac{3}{4}$
6	$12\frac{1}{2}$	$1\frac{7}{16}$	$10\frac{5}{8}$	12	$\frac{3}{4}$
7	14	$1\frac{1}{2}$	$11\frac{7}{8}$	12	$\frac{7}{8}$
8	15	$1\frac{5}{8}$	13	12	$\frac{7}{8}$
9	16	$1\frac{3}{4}$	14	12	$\frac{7}{8}$
10	$17\frac{1}{2}$	$1\frac{7}{8}$	$15\frac{1}{4}$	16	$\frac{7}{8}$
12	20	2	$17\frac{3}{4}$	16	$\frac{7}{8}$
14	$22\frac{1}{2}$	$2\frac{1}{8}$	20	20	$\frac{7}{8}$
15	$23\frac{1}{2}$	$2\frac{3}{16}$	21	20	1
16	25	$2\frac{1}{4}$	$22\frac{1}{2}$	20	1
18	27	$2\frac{3}{8}$	$24\frac{1}{2}$	24	1
20	$29\frac{1}{2}$	$2\frac{1}{2}$	$26\frac{3}{4}$	24	$1\frac{1}{8}$
22	$31\frac{1}{2}$	$2\frac{5}{8}$	$28\frac{3}{4}$	28	$1\frac{1}{8}$
24	34	$2\frac{3}{4}$	$31\frac{1}{4}$	28	$1\frac{1}{8}$

TABLE 80.

LOSS OF HEAT FROM BARE STEAM PIPE.*

Still Air.

Authority of Test.	Descriptive References.	Size of Pipe.	Square Feet of Heating Surface.	Steam Pressure, Gauge.	Steam Temperature.	Temperature of Air.	Difference of Temperature.	Pounds of Steam Condensed per Sq. Ft. per Hour.	B.T.U. per Sq. Ft. per Hr. per Deg. Dif. in Temp.
Barrus..........	Power, Dec., 1901; Trans. A.S.M.E., vol. XXIII; Stevens Ind., Vol. XIX, p. 388.	2	63.57	82	325	56	268.6	0.915	3.01
Do..........		2	63.92	149	365	63.3	302.2	1.150	3.25
Do..........		10	98.33	149	365	73.6	291.7	1.085	3.18
Hudson Beare.	....	3.53†	8.13	135	358	67	291	1.050	3.10
"130 lbs.".....	Stevens Ind., Vol. XIX, p. 388.	2	50.66	128	354	80.1	274.6	0.994	3.13
Jacobus.......	Stevens Ind., Vol. XVIII.	2	7.63	53	301	71.2	229.6	0.707	2.78
Brill...........	Trans. A.S.M.E., Vol. XVI.	8	135.4	110	344	75.5	269	0.834	2.71

* C. P. Paulding, *Stevens Indicator*, Vol. XIX, p. 388. † Outside diameter.

331. Coverings. — Steam pipes, feed-water pipes, boiler steam drums, receivers, separators, etc., should be covered with heat-insulating material to reduce radiation losses to a minimum. For most practical purposes the loss of heat from a bare steam pipe or drum may be taken as 3 B.T.U. per square foot per hour per degree difference in temperature, Table 80. The actual loss depends upon the diameter of the pipe, on its position whether vertical or horizontal, the nature of the surface, and the velocity of the surrounding air currents. For a detailed analysis of these various influences, and interesting information on the transmission of heat, the reader is referred to Paulding's "Steam in Covered and Bare Pipes."

By properly applying any good commercial covering, from 75 per cent to 90 per cent of the heat loss may be prevented. (See Fig. 344 and Table 81.)

FIG. 344. Efficiency of Pipe Coverings.

Example: Required the saving per annum due to covering a pipe 10 inches in diameter and 100 feet long; steam pressure 150 pounds; average temperature of the air 76 degrees F.; cost of covering applied 65 cents per running foot; efficiency of covering 85 per cent; cost of coal $2.50 per ton; plant to operate 14 hours per day and 300 days per year.

The temperature of steam at 150 pounds pressure = 366 degrees F.

Difference of temperature between the steam and air = 366 − 76 = 290 degrees F.

Loss per square foot per hour, bare pipe = 3 × 290 = 870 B.T.U.

Loss per square foot per day, bare pipe = 870 × 14 = 12,180 B.T.U.

Loss per square foot per year, bare pipe = 12,180 × 300 = 3,654,000 B.T.U.

100 lineal feet of 10-inch pipe has an external surface of 282 square feet. Therefore the loss per year from the bare pipe is

282 × 3,654,000 = 1,030,000,000 B.T.U. (approx.).

TABLE 81.

EXPERIMENTS ON STEAM-PIPE COVERINGS.

("Condensation of Steam in Covered and Bare Pipes" [Paulding].)

Kind of Covering.	Diam. of Test Pipe, Inches.	Thickness of Covering, Inches.	Temperatures F.		B.T.U. per Hour per Square Foot of Pipe Surface.		Date of Test.	Testing Expert.
			Steam.	Air.	Total.	Per Degree Difference.		
Hair felt	2	0.96	302.8	71.4	89.6	0.387	1901	Jacobus
Do	8	0.82	348.3	69.0	117.9	0.422	1894	Brill
Remanit for intermediate pressure.	2	0.88	304.5	73.3	100.3	0.434	1901	Jacobus
Remanit for high pressure.	2	1.30	306.6	76.1	83.7	0.363	1901	Jacobus
Mineral wool	8	1.30	344.1	58.3	81.3	0.284	1894	Brill
Champion mineral wool	8	1.44	346.1	74.3	86.1	0.317	1894	Brill
Rock wool	8	1.60	344.1	63.0	72.0	0.256	1894	Brill
Asbestos sponge felted	2	1.125	364.8	60.7	145.0	0.477	1901	Barrus
Do	10	1.375	364.8	62.8	85.0	0.248	1901	Barrus
Do	2	1.14	309.2	79.4	59.7	0.260	1901	Jacobus
Magnesia	4	1.12	388.0	72.0	147.0	0.465	1896	Norton
Do	2	1.09	354.7	80.1	155.8	0.567	1896	Paulding
Do	8	1.25	344.1	66.3	106.6	0.384	1895	Brill
Do	2	1.08	310.9	81.6	69.8	0.304	1901	Jacobus
Do	2	1.00	365.2	64.6	155.0	0.515	1901	Barrus
Do	10	1.19	365.2	66.0	103.0	0.347	1901	Barrus
Asbestos, Navy Brand	2	1.20	309.2	79.4	69.9	0.304	1901	Jacobus
Do	2	1.125	365.2	64.6	176.0	0.585	1901	Barrus
Do	10	1.375	365.2	66.8	112.0	0.375	1901	Barrus
Manville sectional	8	1.70	345.5	78.3	93.4	0.394	1894	Brill
Do	2	1.31	354.7	80.1	157.0	0.572	1896	Paulding
Do	4	1.25	388.0	72.0	143.0	0.453	1896	Norton
Asbestos air cell	4	1.12	388.0	72.0	166.0	0.525	1896	Norton
Do	2	0.96	303.3	72.3	165.5	0.716	1901	Jacobus
Asbestos fire felt	8	1.30	344.7	79.0	133.5	0.502	1894	Brill
Do	2	1.00	354.7	80.1	198.0	0.721	1896	Paulding
Do	2	0.99	307.4	72.5	180.0	0.766	1901	Jacobus
Fossil meal	8	0.75	347.1	75.3	238.0	0.876	1894	Brill
Riley cement	8	0.75	347.9	74.3	260.0	0.950	1894	Brill

Assuming a net available heat value of 10,000 B.T.U. per pound for the coal, the equivalent coal consumption is 51.5 tons, valued at $51.5 \times \$2.50 = \$128.75$.

The covering will save 85 per cent of this, or $109.50 per annum.

The pipe covering applied will cost $100 \times 0.65 = \$65.00$.

In this case the covering will pay for itself in considerably less than a year.

Pipe covering is applied in sections molded to the required form and held to the pipe by bands, or may be applied in a plastic form. The former is more readily applied and removed, and is usually adopted for pipes, while the valves and fittings are sometimes covered with plastic material. Piping should be tested under pressure before being covered, since leaks destroy the efficiency and life of the covering. If the surrounding atmosphere is moist the covering should be given two or three coats of good paint. Coverings are sometimes applied to cold-water pipe to prevent sweating in a humid atmosphere.

Pipe Coverings: Power, July, 1904, p. 407, Aug., 1904, p. 482, May, 1903, p. 239, Dec., 1901, p. 32; St. Ry. Jour., Nov. 29, 1902, p. 875; Engr., Lond., May 27, 1904, p. 547; Eng. Review, Nov., 1898, p. 15; Am. Elecn., May, 1903; Engng., Aug. 7, 1903; Mech. Engr., Nov. 25, 1905; Elec. World and Engr., April 6, 1901; Stevens Ind., Oct., 1902; Trans. A.S.M.E., 16–827, 23–791.

Identification of Power House Piping by Colors: Power & Engr., Apr. 26, 1910, p. 752.

332. Expansion. — One of the most difficult problems in the design of a piping system is the proper provision for expansion and contraction due to change in temperature. If a pipe is immovably fixed at both ends and under no strain when cold, and the temperature is increased, as by the admission of steam, it is subjected to a compression proportional to the rise in temperature (within the elastic limit). For example, a 6-inch standard extra heavy wrought-iron pipe 200 feet long at 66 degrees F., if heated to 366 degrees F. (the temperature corresponding to steam at 165 pounds per square inch absolute pressure), will exert an axial force of

$$P = EA\,(t_1 - t)\mu. \quad (\textit{Mechanics of Engng.}, \text{Church, p. 218.}) \qquad (147)$$

P = force in pounds.

E = modulus of elasticity, 30,000,000.

t_1 = final temperature, degrees F.

t = initial temperature.

μ = coefficient of expansion, 0.0000075.

A = sectional area of the pipe material, 8.5 square inches.

Hence

$$P = 30{,}000{,}000 \times 8.5\,(366 - 66)\,0.0000075.$$
$$= 573{,}750 \text{ pounds.}$$

Unless well braced throughout its entire length the pipe will buckle and become distorted. If free to expand its length would increase. The temperature of the pipe is always less than that of the steam on account of radiation from the outer surface and varies with the efficiency of the covering. But ignoring radiation the increase in length is

$$l = \mu (t_1 - t) L, \tag{148}$$

in which

l = increase in length, inches.
L = length of pipe, inches.
Other notations as in (147).

Substituting in (148), $t_1 = 366$.

$t = 66$.
$\mu = 0.0000075$.
$L = 2400$.
$l = 0.0000075\,(366 - 66)\,2400$
$= 5.4$ inches.

Since the forces produced by expansion are practically irresistible, the pipe is invariably allowed to expand freely by suitable means so as not to strain the connections. The coefficients of expansion per degree difference in temperature for various pipe materials are given in Table 82.

Headers less than 50 feet in length usually require no special provisions for expansion provided the ends are free and the leads to and

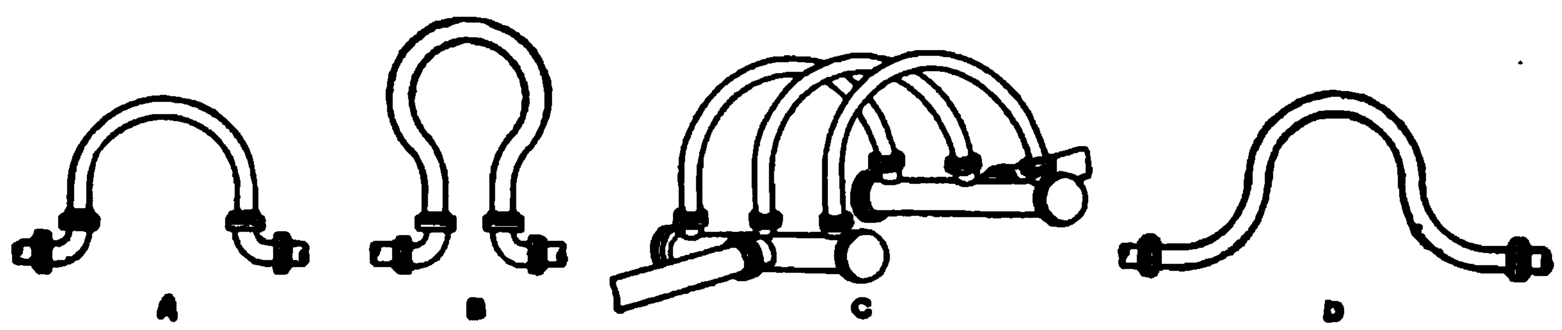


FIG. 345. Types of Expansion Pipe Bends.

from the header are not too short, the pipe usually being anchored at the middle and permitted to expand in either direction. Free expansion of the feeders may be provided for

(1) By *long radius bends*, as in Fig. 345.
(2) By *double swing screwed fittings*, as in Fig. 346, or
(3) By *packed expansion joints*, Fig. 347.

Where practicable the long radius bends will prove most satisfactory. The radius of the bend should not be less than 5 diameters of the pipe, and larger if possible. The length of straight pipe at the

end of each bend should not be less than twice the diameter of the pipe measured from the face of the flange.

On account of the great strains to which the joints of pipe bends are subjected, the welded joint, *G*, Fig. 343, is recommended as giving the best results. The next best is the lap joint, *F*, Fig. 343.

TABLE 82.

COEFFICIENTS OF LINEAR EXPANSION PIPING MATERIALS.

Material.	Temperature Range.	Mean Coefficient per Degree F.
Wrought iron and mild steel	32–212	0.00000656
Wrought iron	32–572	0.00000895
Cast iron	32–212	0.00000618
Cast steel	32–212	0.00000600
Hardened steel	32–212	0.00000689
Nickel-steel, 36 per cent Nickel	32–572	0.00000030
Copper, cast	32–212	0.00000955
Copper, wrought	32–572	0.00001092
Lead	32–212	0.00001580
Cast brass	32–212	0.00001043
Brass wire and sheets	32–212	0.00001075
Tin cast	32–212	0.00001207
Tin hammered	32–212	0.00001500
Zinc cast	32–212	0.00001633
Zinc hammered	32–212	0.00001722

LINEAR EXPANSION OR CONTRACTION OF CAST IRON IN INCHES PER 100 FEET, — DEGREES F.

Temperature Difference.	Expansion.	Temperature Difference.	Expansion.
100	0.72	300	2.376
150	1.1016	400	3.360
200	1.5024	500	4.440
250	1.9260	600	5.616
....	....	800	7.872

Multiply by 1.1 for wrought mild steel.
Multiply by 1.5 for wrought copper.
Multiply by 1.6 for wrought brass.

Fig. 345, *A*, *B*, *C*, *D* shows applications of pipe bends to straight pipe runs. *A* is the cheapest and most common arrangement for all sizes of pipe. *B* is a modification for limited center to center spaces. *C* shows a common method of taking up expansion in straight runs of pipe of very large diameters where the space requirements prohibit the use of

a single U bend. Here the main runs are connected to manifolds which in turn are connected by a number of small U bends, the equivalent areas of which correspond to that of the large pipes. This makes a more flexible connection than if a single U bend were used. The arrangement D does away with the elbows required in A, but is not applicable to pipes over 8 inches in diameter.

Figs. 357 and 358 show applications of pipe bends to boiler and header connections.

Fig. 346 shows a double swing screwed joint in which expansion causes the fittings to turn slightly and thus relieve the strain. This method is usually adopted where long radius bends are not practicable on account of lack of space and where screwed fittings are used.

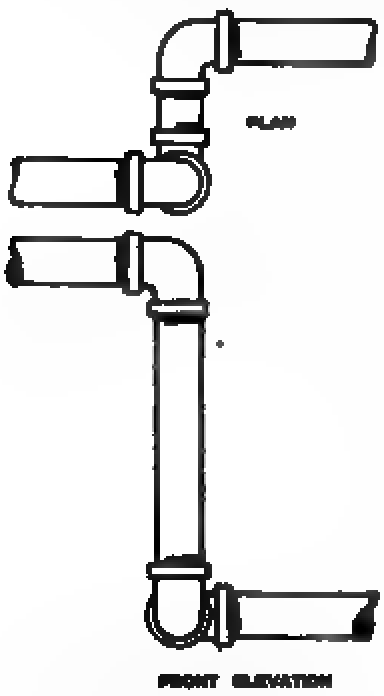


FIG. 346. "Double-Swing" Expansion Joint.

FIG. 347. Slip Expansion Joint.

Slip joints, Fig. 347, are now little used except with very large pipes and where space prohibits long radius bends. When slip joints are employed the pipe must be securely anchored to prevent the steam pressure from forcing the joint apart and at the same time permit the pipe in expanding to work freely in the stuffing box. Sagging of the pipe on either side, which might cause binding in the joint, is prevented by suitable supports.

Expansion in Steam Pipes: Power, July, 1906, p. 426, Jan., 1904, p. 30, March, 1904, p. 160, Oct., 1904, p. 609, Dec., 1900; Am. Elecn., 10–432; Engr., U.S., Feb. 1, 1904, p. 125; Eng. News, 44–194, 47–468, 50–487; Power, June 2, 1908.

333. Pipe Supports and Anchors. — Pipe lines must be supported to guard against excessive deflection and vibration. Supports are conveniently classified as (1) hangers, (2) wall brackets, and (3) floor stands.

Fig. 348 illustrates a type of hanger for suspending pipes from I beams. The supports being free to swing, no provision for expansion is necessary. A properly designed hanger may be readily removed without disturbing the pipe line, and should be adjustable to facilitate "lining up." If of rigid construction the lower end should be provided with a roller.

Fig. 349 gives the details of a wall bracket with rolls and roll binder.

Supports adjacent to long radius bends should be provided with roll binders as illustrated to prevent the pipe from springing laterally, but

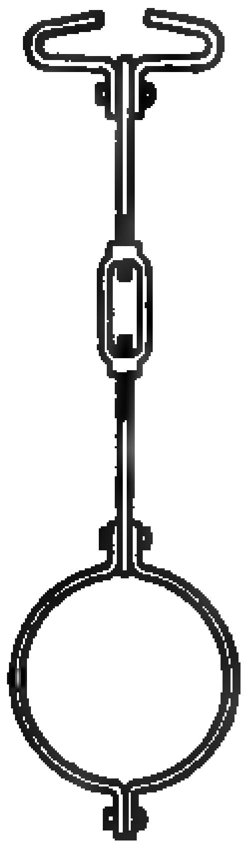


Fig. 348. A Typical Pipe Hanger.

Fig. 349. A Typical Wall Bracket with Binding Roll.

Fig. 349a. A Typical Floor Stand.

they may otherwise be omitted. The rollers are often made adjustable to facilitate lining up.

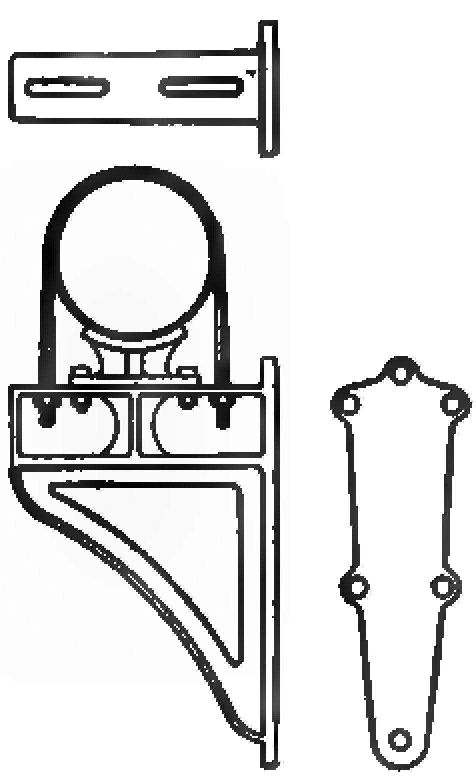


Fig. 350. A Typical Pipe Anchor.

Fig. 349a illustrates a typical floor stand. Pipe lines are usually securely anchored at suitable points in a manner similar to that illustrated in Fig. 350, the pipe resting on a saddle and being rigidly clamped to the bracket by a flat iron band with ends threaded and bolted. This limits expansion to one direction and prevents excessive strain on the fittings.

334. General Arrangement of High-Pressure Steam Piping. — The general arrangement of piping depends in a great measure upon the space available for engines and boilers.

The engine and boiler room may be placed

(1) Back to back, Fig. 361, (2) End to end, Fig. 351,
(3) Double decked, Fig. 356.

The *back to back* arrangement is the most common and, other things permitting, is to be preferred on account of the short and direct connection between engines and boilers and the ease of enlargement. The engine and boiler rooms are separated by a wall, and as much of the piping as possible is located in the boiler room.

The *end to end* arrangement is ordinarily limited to situations where the distribution of space precludes the back to back system.

The *double decked* arrangement is frequently used where ground space is limited or expensive.

Engines and boilers are connected in a variety of ways through steam headers as shown in the following examples:

(1) Single header, Fig. 351,
(2) Duplicate headers, Fig. 352,
(3) Loop or ring header, Fig. 353,
(4) The "unit" system, Fig. 357.

The *single header* system is perhaps the most common, since it embodies simplicity, low first cost, and provision for extension.

The *duplicate* system is losing favor, since experience shows that the extra cost of the duplicate mains will usually give better returns in continuity of operation and maintenance if invested in high-grade fittings on a single pipe system.

The *loop header* is well adapted for the power plants of tall office build-

FIG. 351. Plan of High Pressure Piping, Princeton University Power Plant.

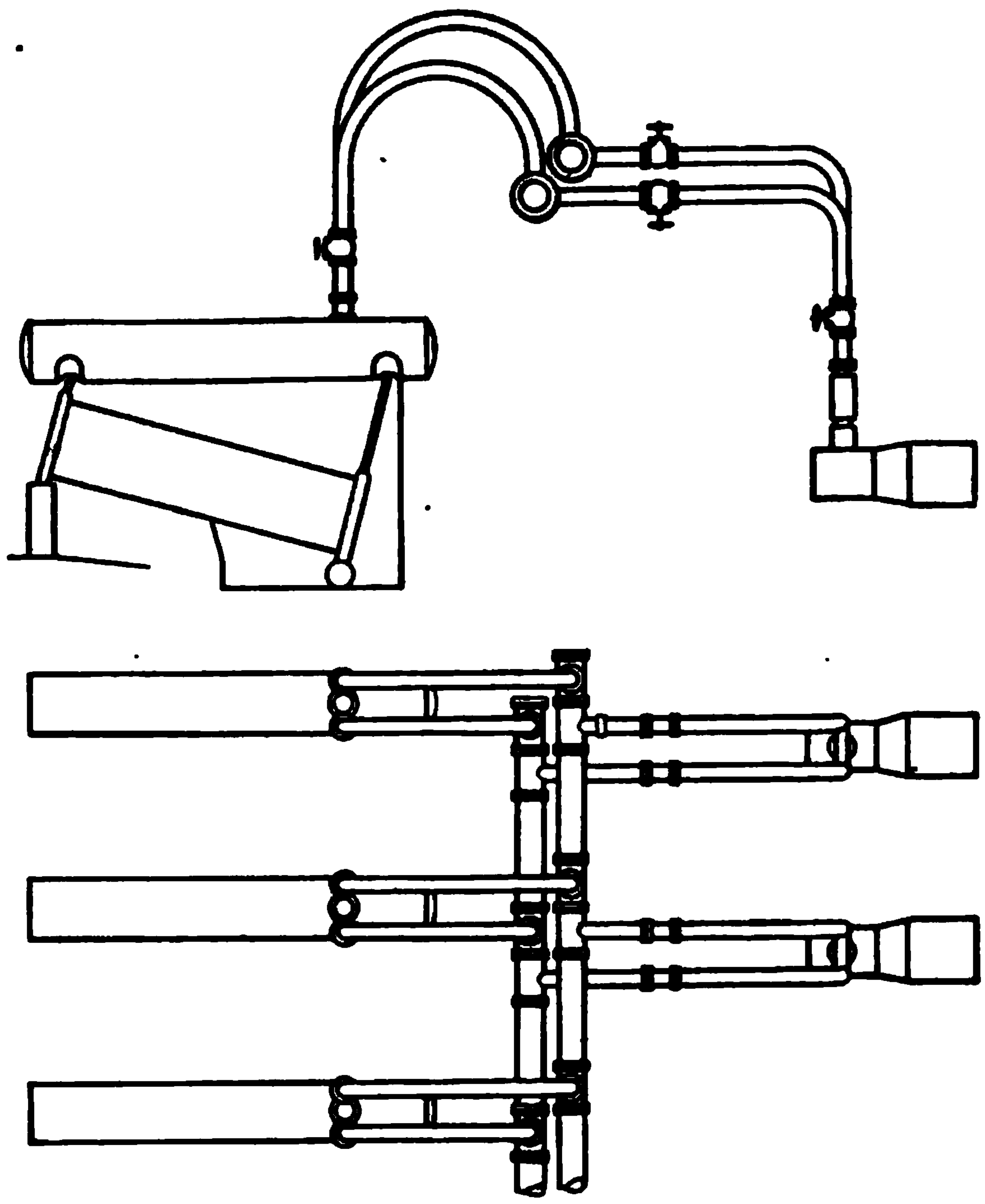


FIG. 352. Typical Duplicate Header System.

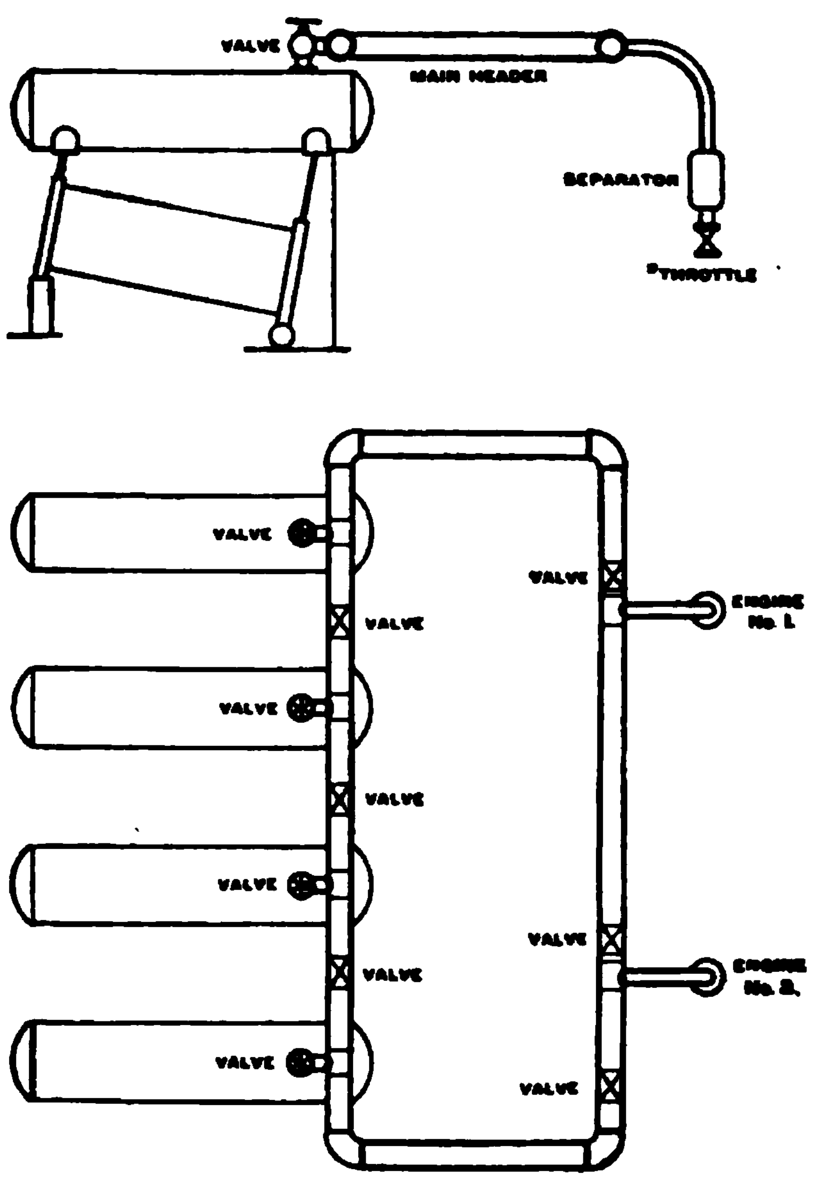


FIG. 353. Typical Loop Header.

ings, Fig. 355, in which a large number of steam engines, elevator pumps, air compressors, and miscellaneous steam-consuming appliances are crowded together in a comparatively small space.

Large modern power plants are, by the latest practice, divided into complete and independent units, as in Fig. 440, each prime mover having its own boiler equipment, coal and ash-handling machinery, feed pumps, and piping, operated independently of the rest of the plant, though provision is made whereby any boiler equipment may provide steam for any prime mover.

FIG. 354. Typical By-Pass System.

The power plant of the Manhattan Elevated Railway Company, New York, is practically divided into eight sections each consisting of an engine and eight boilers, the boilers being "double decked" (Fig. 356).

The branch pipes from the upper and lower batteries lead into 18-inch headers, the steam from each being conducted to a receiver reservoir 36 inches in diameter and 20 feet long in the engine room basement directly behind each engine, from which the two high-pressure cylinders are supplied. Gate valves are used in each boiler branch, one close to the boiler and another near the header, and also in the steam pipes near the reservoir. The steam headers for each of the eight units are connected by a main which equalizes the pressure and allows a deficiency in one unit to be made up from the others.

Figs. 357 and 358 show the general arrangement of the steam piping at the Yonkers power house of the New York Central. The turbines

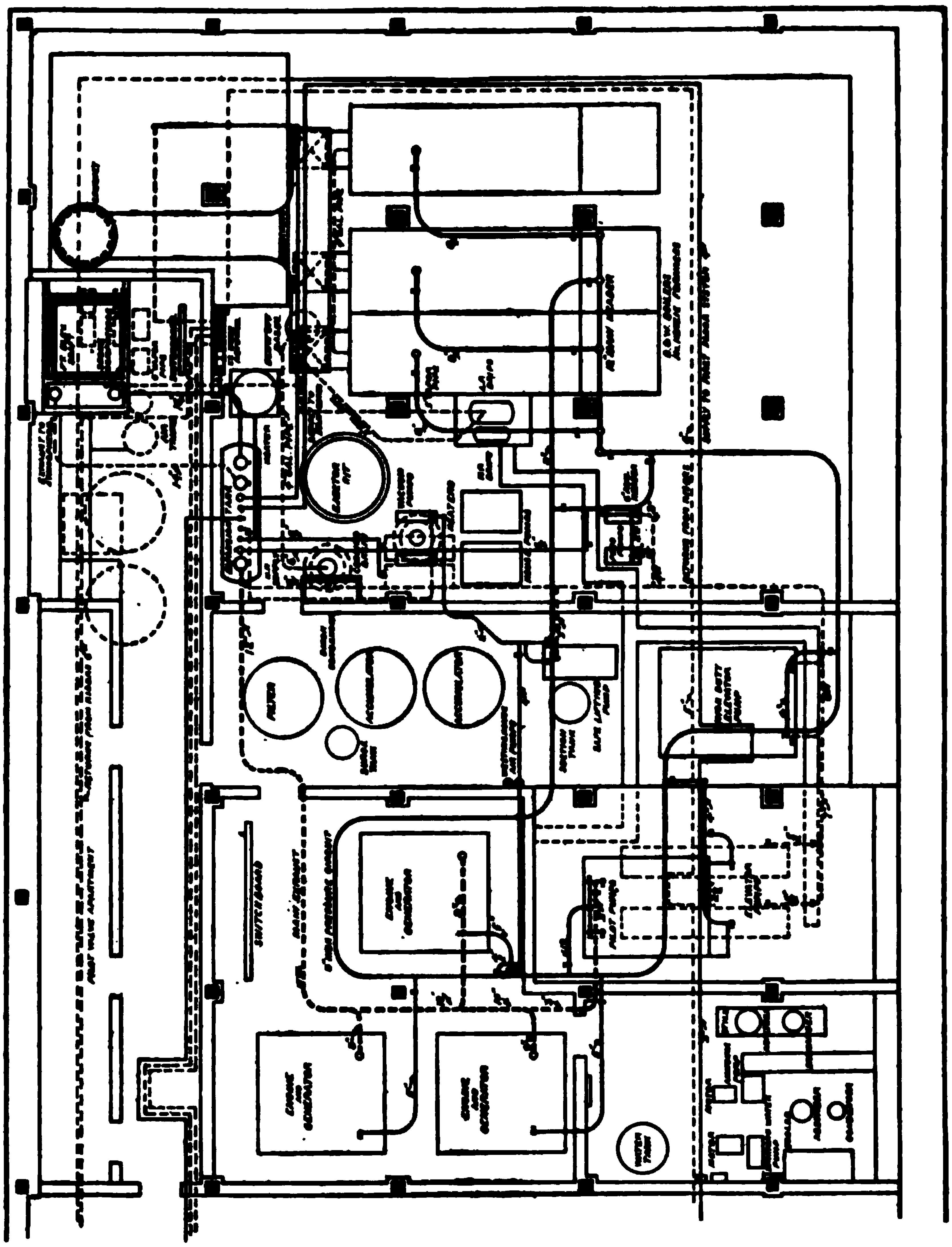


FIG. 355. General Arrangement of Steam and Exhaust Piping, Heyworth Building, Chicago, Ill.

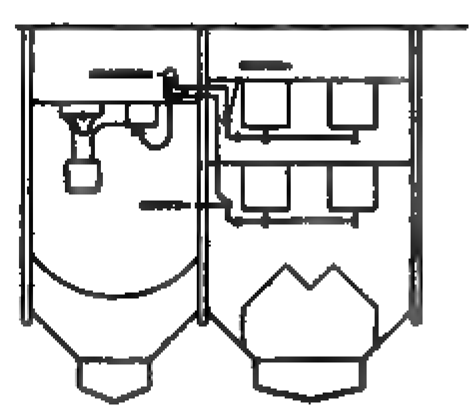


Fig. 356. General Arrangement of Steam Piping, Manhattan Elevated Station.

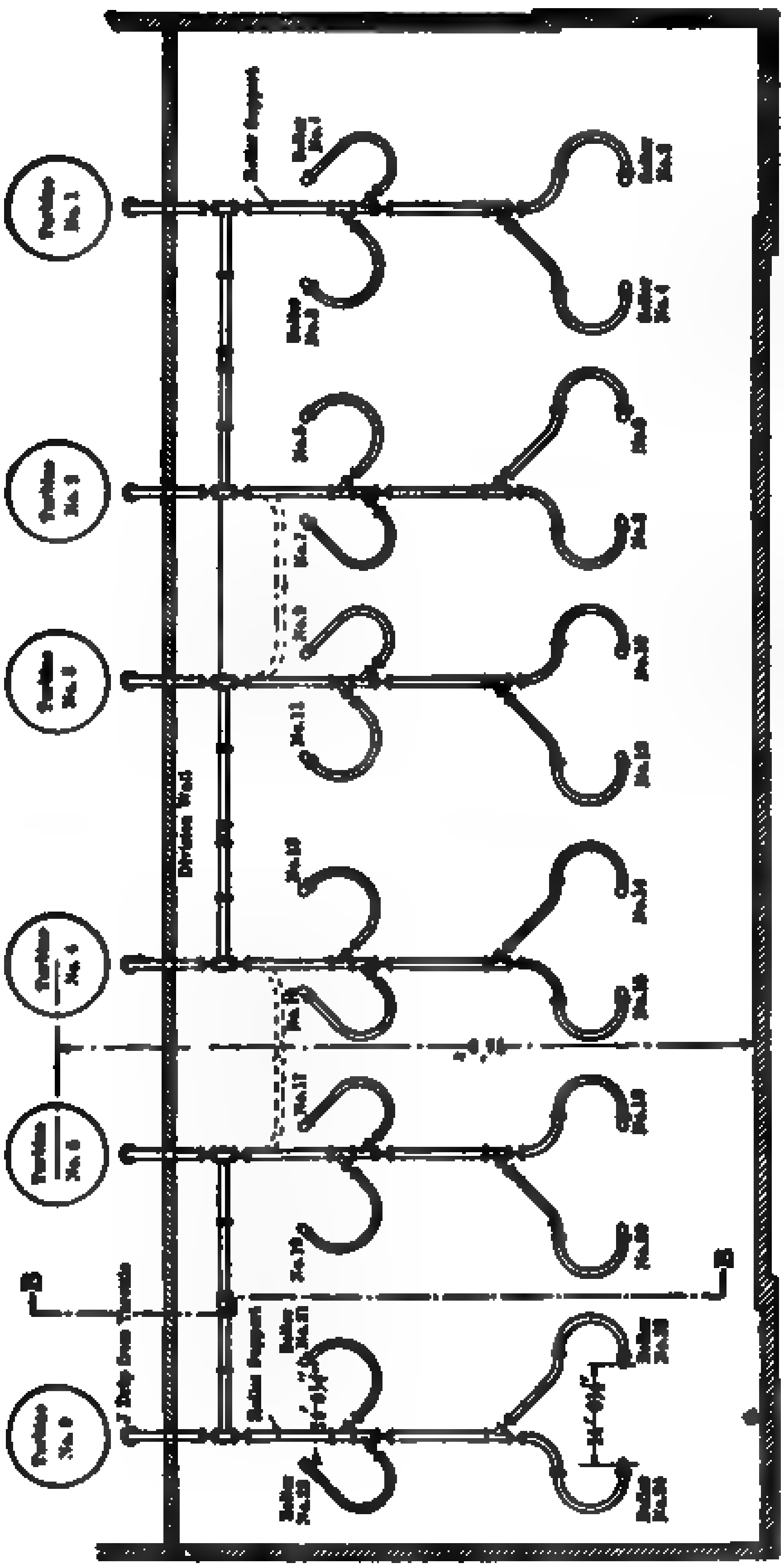


Fig. 357. Plan of High Pressure Piping, Yonkers Power House of New York Central R.R.

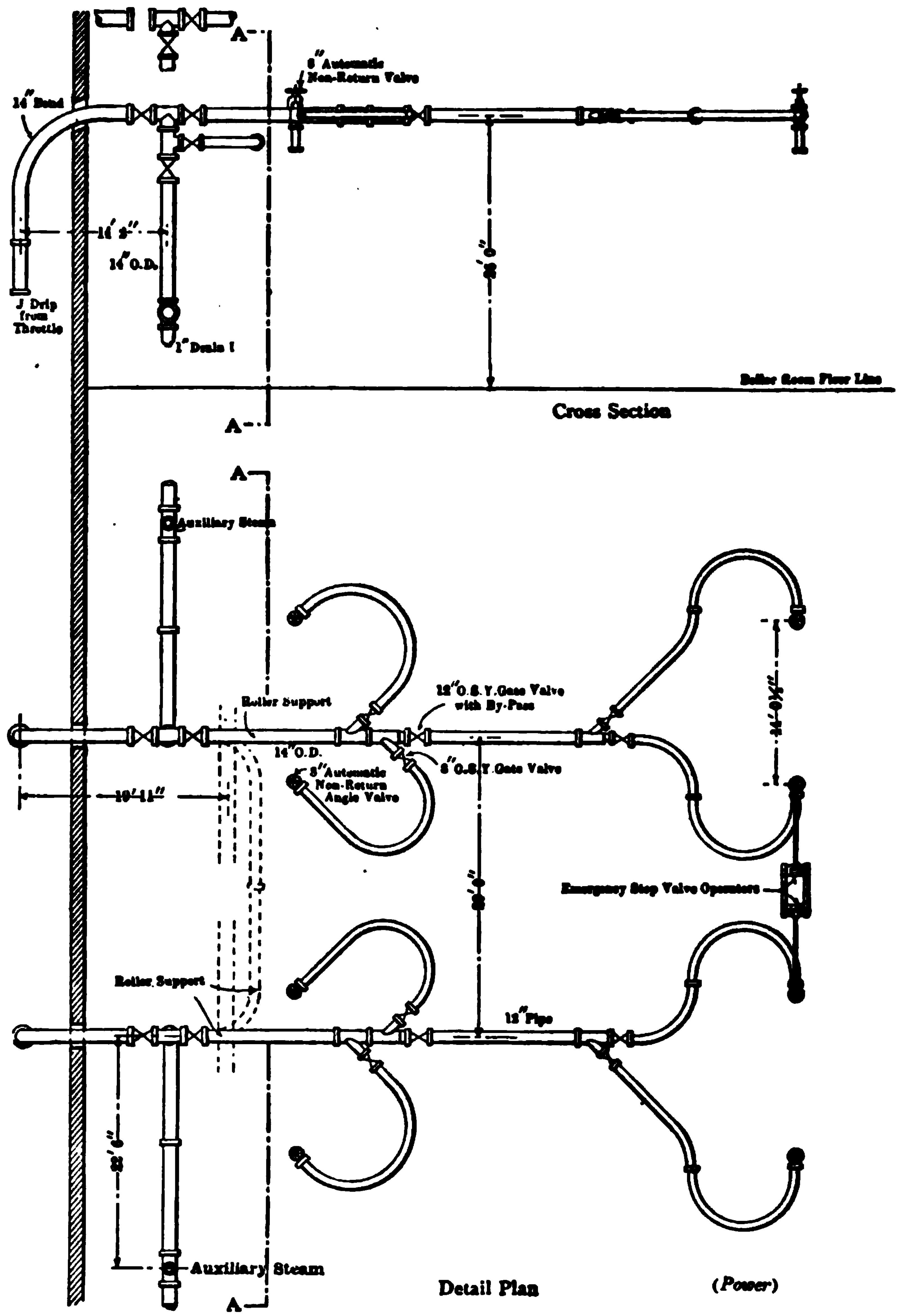


FIG. 358. Details of Boiler Steam Piping, Yonkers Power House of the New York Central R.R.

are connected in pairs by 14-inch loops, each turbine taking steam from either of two banks of four boilers. The high-pressure steam piping is of mild steel with modified reënforced "Van Stone" joints. The high-pressure valves are of the split disk pattern with semi-steel bodies. Expansion is taken up by the long sweep bends.

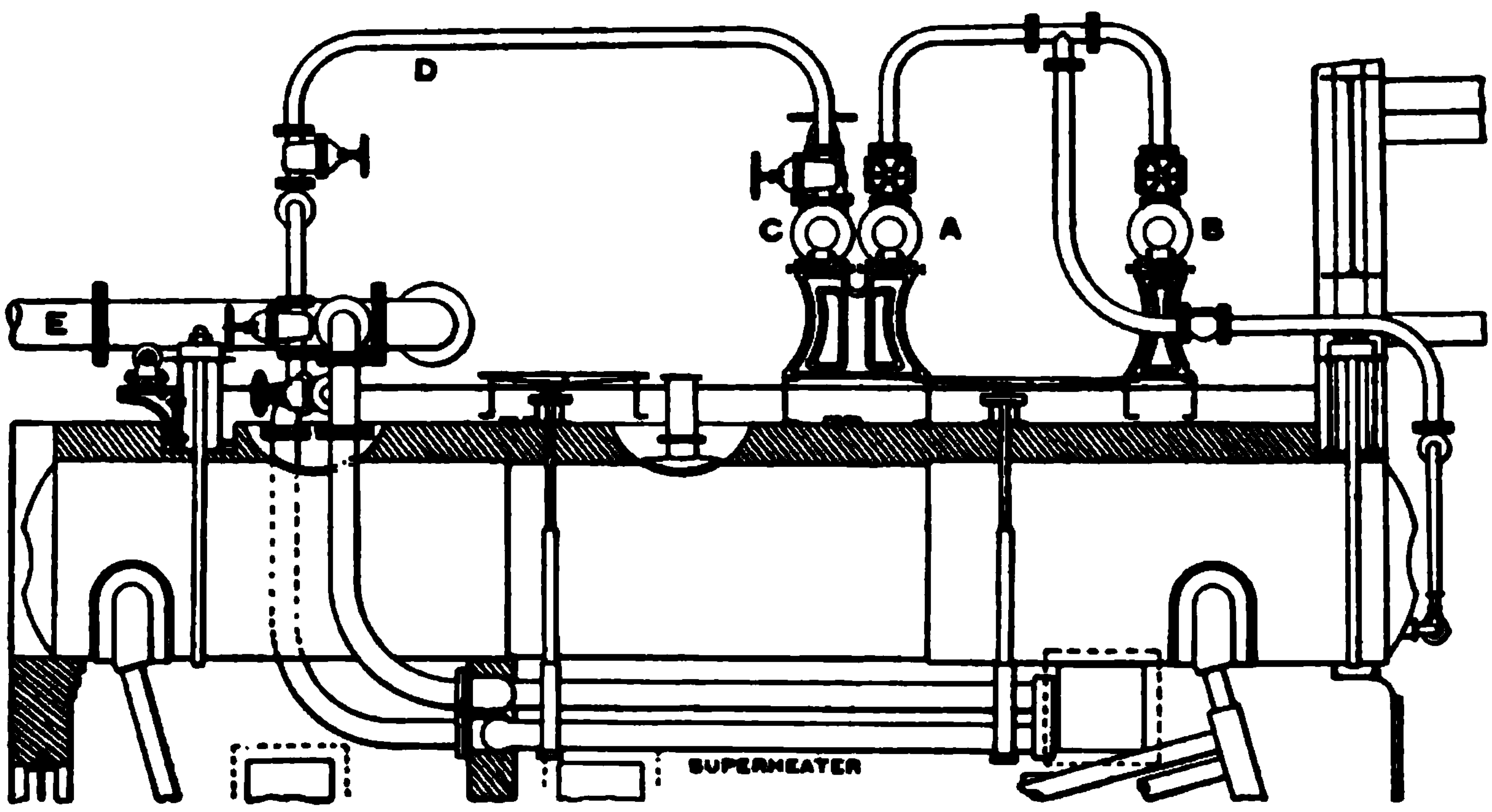


FIG. 359. Overhead Boiler Piping, Quincy Point Power Plant of the Old Colony St. Ry. Co., Quincy Point, Mass.

Plants using superheated steam are ordinarily piped to supply saturated steam to the auxiliaries as illustrated in Fig. 359. The boiler branch *E*, leading to the main header, normally supplies superheated steam to the engines. *C* is an auxiliary main supplying the air pumps, stoker engines, and other auxiliaries with saturated steam from branch pipe *D*.

Steam Piping: Power, Feb. 23, 1909, Nov., 1905, p. 683, April, 1904, p. 213, Feb., 1904, p. 90, Sept., 1904, p. 540, Nov., 1904, p. 677, July, 1903, p. 356; Ir. Tr. Rev., May 29, 1902; Ir. Age, Dec. 11, 1902; Met. Work, Feb. 14, 1903; St. Ry. Rev., Nov. 20, 1904, p. 869; St. Ry. Jour., Oct. 15, 1904, p. 441; Am. Elecn., March, 1905, p. 127; Elec. Engr., Lond., Dec. 23, 1904; Engr. U.S., Feb. 15, 1905, Dec. 1, 1904; Elecn., Lond., July 21, 1899; Eng. Rec., 48-90; Trans. A.S.M.E., 15-536; Cass. Mag., June, 1906.

335. Main Steam Headers. — Until quite recently it was the usual practice to make the area of the steam header equivalent to the combined areas of the feeders, but the function of the header is now regarded as that of an equalizer rather than a storage reservoir. In the various large power houses recently built in New York City, with ultimate capacities of from 60,000 to 150,000 kilowatts, the largest steam headers are not over 16 inches in diameter. In some recent

designs the pipes leading from the header to the engines are two sizes smaller than called for by the engine builders. In this case large receiver separators two to four times the volume of the high-pressure cylinder are provided near the throttle as in Fig. 356. The pipes between receiver and engine are full size. The object of the arrangement is to give (1) a constant flow of steam, (2) a full supply of steam close to the throttle, and (3) a cushion near the engine for absorbing the shock caused by cut-off. With moderately superheated steam and boiler pressures from 125 to 150 pounds a velocity of 8000 feet per minute is allowed in the header and as high as 9000 feet per minute between header and receiver. With steam turbines velocities as high as 12,000 feet per minute are permissible, provided the pipe is less than 50 feet in length and practically free from sharp bends. Main headers are ordinarily constructed of mild steel, though cast-iron and cast-steel headers are not uncommon. Cast headers permit of fewer joints and are well adapted to situations where a number of branches are closely grouped as in Fig. 361. Cast-iron headers are employed in the Manhattan Elevated Railway power station, New York.

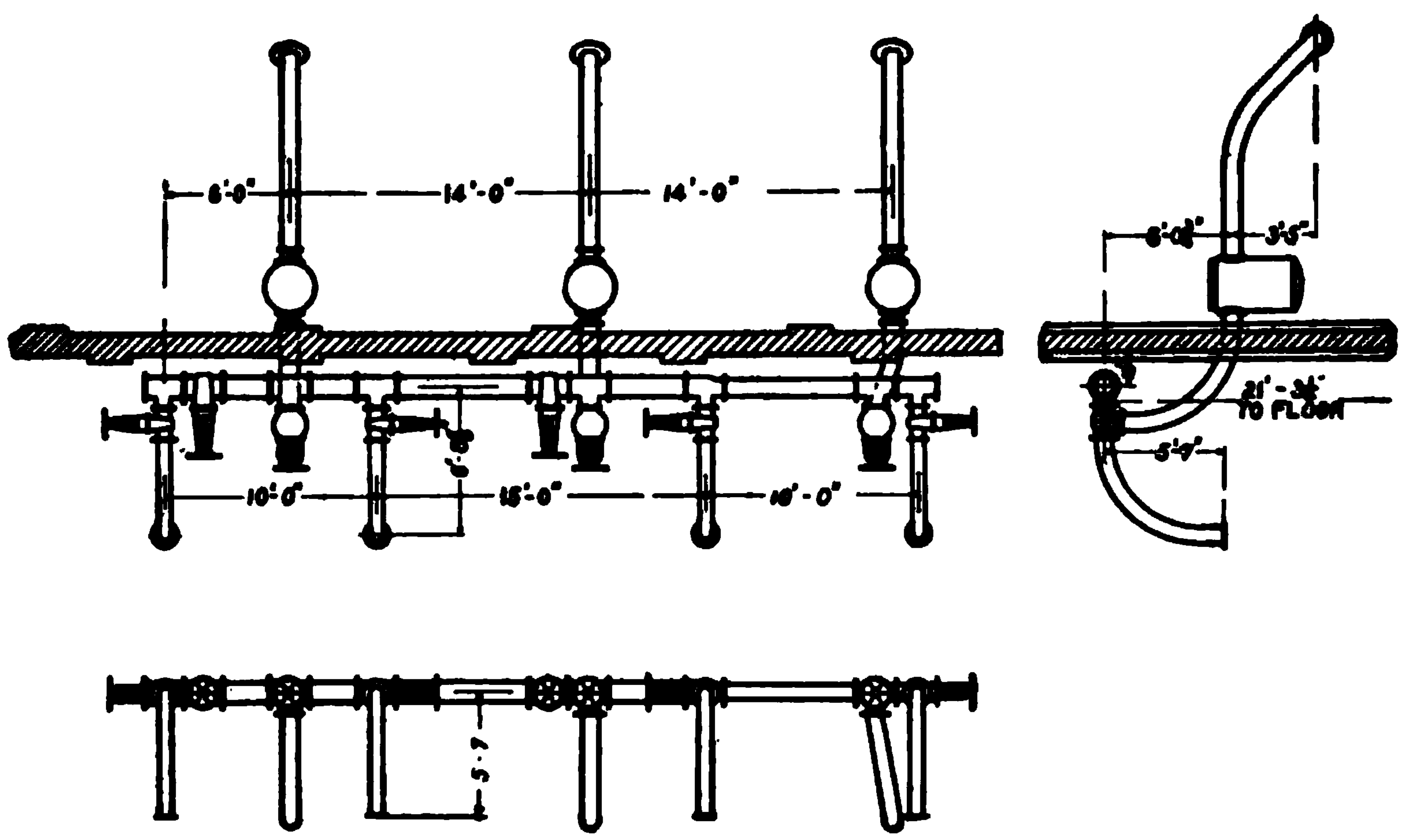


FIG. 360. Steam Header and Branches, Grand Rapids, Grand Haven and Muskegon Ry. Co. Power House.

The proper arrangement and number of valves in the main header and feeders has been a subject of much consideration. Figs. 356 to 361 show some of the different successful arrangements in recent installations. In Fig. 360 there is but one valve between the boiler nozzle and the main header, while in Fig. 361 there are two. The latter is the more common arrangement. Where two valves are

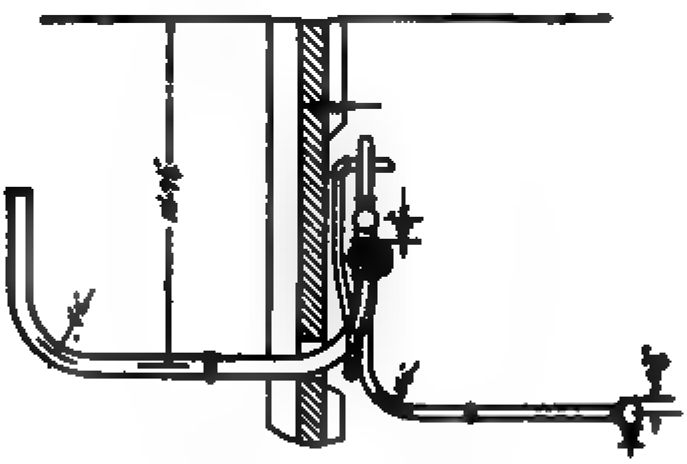


FIG. 361. Steam Header and Branches, Des Moines City Ry. Co.'s Power House

placed in a feeder they should be arranged so as not to form a pocket for the accumulation of leakage. In a number of recent installations, Fig. 358, the valve nearest the boiler is of the "automatic stop and check" type, its function being the automatic cutting off of the steam from the header should the pressure in the boiler suddenly drop as in case of blowing out a tube.

Arrangement of Steam Piping: Am. Elecn., April, 1905, June, 1902, p. 257, June, 1900; Engr. U.S., Dec. 1, 1904; Mech. Engr., Nov. 4, 1905; Power, Sept., 1904, p. 511, July, 1902; Eng. News, Nov. 26, 1903, p. 487; Elec. Rev., Lond., Aug. 11, 1899, p. 251; St. Ry. Rev., Jan., 1900, p. 12; Nov., 1904, p. 869.

336. Flow of Steam in Pipes.* — The several accepted formulas relating to the flow of steam in pipes have been based upon a few experiments limited to pipes of small diameter, hence the application of these formulas to larger pipes or to conditions other than those under which they were deduced is apt to lead to considerable error. In small plants extreme accuracy in determining the proper sizes is not necessary; it is better to err in the installation of too large a pipe than one too small. In larger stations, however, where the pipes are large and the pressure high, the cost of the piping increases very rapidly with the size. For example, the cost of 10-inch high-pressure fittings is from 15 to 20 per cent greater than 9-inch fittings, and in large installations this first cost item may be of considerable importance.

The simplest and most commonly used formula is based upon an allowable steam velocity of 6000 feet per minute, friction and other causes of drop in pressure being disregarded; thus, for a velocity of 6000 feet per minute,

$$d = 0.175\left(\frac{W}{\gamma}\right)^{\frac{1}{2}}, \tag{149}$$

in which

d = diameter of the pipe in inches,
γ = density of the steam in pounds per cubic feet, and
W = weight of steam flowing in pounds per minute.

In determining the diameter of the steam pipe opening for reciprocating engines a much lower velocity than 6000 feet per minute is assumed, to allow for the various conditions of operation. Average practice gives the constant in equation (149) a value of 0.3 instead of 0.175 when used in this connection.

Equation (149) gives satisfactory results for pipes under 100 feet in length and between 4 and 8 inches in diameter; for larger diameters the velocity could be increased with advantage; for smaller diameters

* See author's original paper, *Power*, June, 1907, p. 377.

or greater lengths friction and condensation would cause considerable drop in pressure and some one of the approved formulas in Table 83 should be used instead.

Mean Velocity, Feet per Minute

FIG. 362. Drop in Pressure for Various Velocities and Pipe Sizes. Initial Pressure 106 Pounds Gauge, Length of Pipe 100 feet.

A large drop in pressure means a small pipe and high velocity with consequent decrease in condensation, but a point is soon reached where the economy in the size of pipe is more than offset by the loss in friction. There seems to be no fixed rule for determining the drop

TABLE 83.

FORMULAS FOR THE FLOW OF STEAM IN PIPES.

No. of Equation	Author.	Reference.	Formula as Given by the Author.	Formula Reduced to Equivalent Notations. English Units.*
11	G. H. Babcock	"Steam" (1890). Pub. by B. & W. Co.	$W = 87 \left\{ \frac{yPd^5}{L\left(1 + \frac{3.6}{d}\right)} \right\}^{\frac{1}{2}}$	$W = 87 \left\{ \frac{yPd^5}{L\left(1 + \frac{3.6}{d}\right)} \right\}^{\frac{1}{2}}$
12	R. C. Carpenter	Trans. A.S.M.E., Vol. XX, p. 347	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{yd^5}$	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{yd^5}$
7	Gutermuth	Zeit. d. Ver. D. Ing., April 16, 1904, p. 562.	$P = 0.0015\, y \frac{L}{d} V^2$	$P = 105.8 \cdot 10^{-10} y \frac{L}{d} V$
9	Geipel and Kilgour	London Electrician, May 5, 1893.	$P = \frac{W^2 L}{2525\, yd^5}$	$P = \frac{W^2 L}{2525\, yd^5}$
5	Hawksley	Pro. Inst. C. E., Vol. XXXIII, p. 55.	$V = 48 \left\{ \frac{Hd}{L} \right\}^{\frac{1}{2}}$	$V = 9976 \left\{ \frac{Pd}{yL} \right\}^{\frac{1}{2}}$
15	Hurst	King's "Treatise on Coal Gas," Vol. II, p. 374.	$Q = 1350 \left\{ \frac{D^5 P}{GL} \right\}^{\frac{1}{2}}$	$V = 10,360 \left\{ \frac{Pd}{yL} \right\}^{\frac{1}{2}}$
14	Ledoux	Annales des Mines, Vol. II, 1892.	$d = 0.056 \left\{ \frac{W^2 L}{P_0^{1.95} - P_1^{1.95}} \right\}^{\frac{1}{5}}$	$d = 0.699 \left\{ \frac{W^2 L}{P_1^{1.95} - P_2^{1.95}} \right\}^{\frac{1}{5}}$
9a	Martin	Engineering, March 19, 1897	$P = \frac{W^2 L}{3192\, yd^5}$	$P = \frac{W^2 L}{3192\, yd^5}$
4	Unwin	"Encyclopedia Britannica," Vol. XII, pp. 508, 516.	$H = 0.0027\left(1 + \frac{3}{10\,D}\right)\frac{V^2}{2\,g} \cdot \frac{4\,L}{D}$	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{yd^5}$

* All notations
P_1 = Initial ... square inch absolute.
P_2 = Final ... square inch absolute.
$P = P_1 -$ ... e.

W = Weight of steam flowing, pounds per minute.
L = Length of pipe, feet.
d = Diameter of pipe, inches.
y = Mean density, pounds per cubic foot.

TABLE 84.

COMPARISON OF PIPE FORMULAS FOR THE FLOW OF STEAM.

	Author.	V = Velocity, Feet per Minute.	W = Weight, Pounds per Minute.	P = Drop in Pressure, Pounds per Square Inch.	d = Diameter, Inches.
Group II.	Geipel and Kilgour	$V = 9240\sqrt{\frac{Pd}{yL}}$	$W = 50.3\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003960\,\frac{W^2L}{yd^5}$	$d = 0.2087\sqrt[5]{\frac{W^2L}{Py}}$
	Gutermuth	$V = 9722\sqrt{\frac{Pd}{yL}}$	$W = 53\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003557\,\frac{W^2L}{yd^5}$	$d = 0.2032\sqrt[5]{\frac{W^2L}{Py}}$
	Hawksley ..	$V = 9976\sqrt{\frac{Pd}{yL}}$	$W = 54.4\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003370\,\frac{W^2L}{yd^5}$	$d = 0.2010\sqrt[5]{\frac{W^2L}{Py}}$
	Martin.....	$V = 10,350\sqrt{\frac{Pd}{yL}}$	$W = 56.5\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003133\,\frac{W^2L}{yd^5}$	$d = 0.1990\sqrt[5]{\frac{W^2L}{Py}}$
	Hurst.....	$V = 10,360\sqrt{\frac{Pd}{yL}}$	$W = 56.5\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003126\,\frac{W^2L}{yd^5}$	$d = 0.1990\sqrt[5]{\frac{W^2L}{Py}}$
Group I.	Babcock ..	$V = 15,950\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001321\,\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	....
	Unwin......	$V = 16,050\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87.5\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001306\,\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	....
	Carpenter ...	$V = 16,050\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87.5\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001306\,\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	....
	Ledoux...	$V = 442$	$W = 2.44$	$P_1^{1.94} - P_2^{1.94} = 0.1660\,\frac{W^2L}{d^5}$	$d = 0.690\sqrt[5]{\frac{W^2L}{P_1^{1.94} - P_2^{1.94}}}$

most suitable for any given set of conditions. In current practice the drop in pressure between boiler and engine ranges from a fraction of one pound to four pounds per square inch per 100 feet of pipe, with an average between one and two pounds.

Table 83 gives a few of the best known formulas for the flow of steam, and Table 84 a comparison between them with respect to velocity, weight discharged, diameter, and the drop in pressure.

Formula 11, Table 83, is the most commonly accepted, and the curves in Fig. 362 are based upon it, assuming a steam pressure of 100 pounds absolute and pipe lengths of 100 feet. Within the limit of 12,000 feet per minute velocity and 10 pounds per square inch drop in pressure the curves are sufficiently accurate for all practical purposes, but beyond this range the results are purely conjectural and may not be accurate, as no recorded experiments have been conducted at these high velocities or with pipes of large diameters.

Though applicable directly to pipes 100 feet long with mean pressure of 100 pounds per square inch absolute, they may be used for any length or pressure. For example, for any length other than 100 feet, multiply the drop given in the curves by the required length in feet and divide by 100. For any pressure other than 100 pounds absolute, multiply the drop given in the curves by 0.2271 (density of steam in pounds) and divide by the density of steam at the required pressure.

Table 85 is the table ordinarily used in connection with the flow of steam and is calculated from equation 11. Table 86 is based upon equations 4 to 12. The results differ slightly from those in Table 85, though the latter is more comprehensive. The left-hand half of Table 86 gives the discharge in pounds per minute for pipes of various diameters corresponding to drop of pressure as given on the right-hand side in the same horizontal line; e.g., a 6-inch pipe 100 feet long discharges 371 pounds of steam per minute for a drop of 16.4 pounds at 100 pounds pressure.

337. Equation of Pipes. — It is frequently desirable to know what number of one sized pipes will be equal in capacity to another pipe.

According to the formulas in Group II, Table 84, the weights discharged vary with the square root of the fifth power of the diameter; that is, the number of pipes equal in capacity to any given pipe may be determined from the equation

$$N_1 = d^{\frac{5}{2}} \div d_1^{\frac{5}{2}}, \tag{150}$$

in which N_1 = number of pipes of diameter d_1 equal in capacity to a pipe of diameter d; d_1 and d in inches.

TABLE 85.

FLOW OF STEAM THROUGH PIPES (BABCOCK).

Initial Pressure by Gauge. Pounds per Square Inch.	Diameter of Pipe, in Inches. Length of each = 240 diameters.						
	¾	1	1½	2	2½	3	4
	Weight of Steam per Minute, in pounds, with One Pound Loss of Pressure.						
1	1.16	2.07	5.7	10.27	15.45	25.38	46.85
10	1.44	2.57	7.1	12.72	19.15	31.45	58.05
20	1.70	3.02	8.3	14.94	22.49	36.94	68.20
30	1.91	3.40	9.4	16.84	25.35	41.63	76.84
40	2.10	3.74	10.3	18.51	27.87	45.77	84.49
50	2.27	4.04	11.2	20.01	30.13	49.48	91.34
60	2.43	4.32	11.9	21.38	32.19	52.87	97.60
70	2.57	4.58	12.6	22.65	34.10	56.00	103.37
80	2.71	4.82	13.3	23.82	35.87	58.91	108.74
90	2.83	5.04	13.9	24.92	37.52	61.62	113.74
100	2.95	5.25	14.5	25.96	39.07	64.18	118.47
120	3.16	5.63	15.5	27.85	41.93	68.87	127.12
150	3.45	6.14	17.0	30.37	45.72	75.09	138.61

Initial Pressure by Gauge. Pounds per Square Inch.	Diameter of Pipe, in Inches. Length of each = 240 diameters.						
	5	6	8	10	12	15	18
	Weight of Steam per Minute, in Pounds, with One Pound Loss of Pressure.						
1	77.3	115.9	211.4	341.1	502.4	804	1177
10	95.8	143.6	262.0	422.7	622.5	996	1458
20	112.6	168.7	307.8	496.5	731.3	1170	1713
30	126.9	190.1	346.8	559.5	824.1	1318	1930
40	139.5	209.0	381.3	615.3	906.0	1450	2122
50	150.8	226.0	412.2	665.0	979.5	1567	2294
60	161.1	241.5	440.5	710.6	1046.7	1675	2451
70	170.7	255.8	466.5	752.7	1108.5	1774	2596
80	179.5	269.0	490.7	791.7	1166.1	1866	2731
90	187.8	281.4	513.3	828.1	1219.8	1951	2856
100	195.6	293.1	534.6	862.6	1270.1	2032	2975
120	209.9	314.5	573.7	925.6	1363.3	2181	3193
150	228.8	343.0	625.5	1009.2	1486.5	2378	3481

For any other length divide 240 by the given length expressed in diameters and multiply the tabular quantity by the square root of this quotient, which will give the flow for one pound loss of pressure. Conversely, dividing the given length by 240 will give the loss of pressure for the flow given in the table.

TABLE 86.

FLOW OF STEAM THROUGH PIPES (SICKLES).

Length of Pipe One Thousand Feet.

Discharge in Pounds per Minute Corresponding to Drop in Pressure on Right for Pipe Diameters in Inches in Top Line.											Drop in Pressure in Pounds per Square Inch Corresponding to Discharge on Left; Densities and Corresponding Absolute Pressures per Square Inch in First Two Lines.								
Diameter.	12"	10"	8"	6"	4"	3"	2½"	2"	1½"	1"	Density. Pressure.	0.208 90	0.230 100	0.284 125	0.328 150	0.401 180	0.443 200	0.506 230	0.548 250
Discharge	2328	1443	799	371	123.	55.9	28.8	18.1	6.81	2.52	Drop.	18.10	16.4	13.3	11.1	9.39	8.50	7.44	6.87
Do....	2165	1341	742	344	114.6	51.9	27.6	16.8	6.52	2.34	Do..	15.60	14.1	11.4	9.60	8.09	7.33	6.41	5.92
Do....	1996	1237	685	318	106.	47.9	26.4	15.5	6.24	2.16	Do..	13.3	12.0	9.74	8.18	6.90	6.24	5.47	5.05
Do....	1830	1134	628	292	97.0	43.9	25.2	14.2	5.95	1.98	Do..	11.1	10.0	8.13	6.83	5.76	5.21	4.56	4.21
Do....	1663	1031	571	265	88.2	39.9	24.0	12.9	5.67	1.80	Do..	9.25	8.36	6.78	5.69	4.80	4.34	3.80	3.51
Do....	1580	979	542	252	83.8	37.9	22.8	12.3	5.29	1.71	Do..	8.33	7.53	6.10	5.13	4.32	3.91	3.42	3.16
Do....	1497	928	514	239	79.4	35.9	21.6	11.6	5.00	1.62	Do..	7.48	6.76	5.48	4.60	3.88	3.51	3.07	2.84
Do....	1414	876	485	226	75.	33.9	20.4	10.9	4.72	1.53	Do..	6.67	6.03	4.88	4.10	3.46	3.13	2.74	2.53
Do....	1331	825	457	212	70.6	31.9	19.2	10.3	4.43	1.44	Do..	5.91	5.35	4.33	3.64	3.07	2.78	2.43	2.24
Do....	1248	873	428	199	66.2	28.9	18.0	9.68	4.15	1.35	Do..	5.19	4.69	3.80	3.19	2.69	2.44	2.13	1.97
Do....	1164	722	400	186	61.7	27.9	16.8	9.03	3.86	1.26	Do..	4.52	4.09	3.31	2.78	2.34	2.12	1.86	1.72
Do....	1081	670	371	172	57.3	25.9	15.6	8.38	3.68	1.17	Do..	3.90	3.53	2.86	2.40	2.02	1.83	1.60	1.48
Do....	998	619	343	159	52.9	23.9	14.4	7.74	3.40	1.08	Do..	3.32	3.00	2.43	2.04	1.72	1.56	1.36	1.26
Do....	915	567	314	146	48.5	21.9	13.2	7.10	3.11	0.99	Do..	2.79	2.52	2.04	1.72	1.45	1.31	1.15	1.06
Do....	832	516	286	132	44.1	20.0	12.0	6.45	2.83	0.90	Do..	2.31	2.09	1.69	1.42	1.20	1.08	0.949	0.877
Do....	748	464	257	119	39.7	18.0	10.8	5.81	2.55	0.81	Do..	1.87	1.69	1.37	1.15	0.97	0.878	0.769	0.710
Do....	665	412	228	106	35.3	16.0	9.6	5.16	2.26	0.72	Do..	1.47	1.33	1.08	0.905	0.762	0.690	0.604	0.558
Do....	582	361	200	92.8	30.9	14.0	8.4	4.52	1.98	0.63	Do..	1.13	1.02	0.828	0.695	0.586	0.531	0.456	0.429

To get the pressure drop for lengths other than 1000 feet, multiply by lengths in feet + 1000.

According to the formulas in Group I, Table 84, the weights discharged vary as $\left\{d^5 \div \left(1 + \frac{3.6}{d}\right)\right\}^{\frac{1}{2}}$ and the equation becomes

$$N_1 = \left(\frac{d^5}{1 + \frac{3.6}{d}} \div \frac{d_1^5}{1 + \frac{3.6}{d_1}}\right)^{\frac{1}{2}} \tag{151}$$

$$= \left(\frac{d^6 (d_1 + 3.6)}{d_1^6 (d + 3.6)}\right)^{\frac{1}{2}} \tag{152}$$

$$= \frac{d^3 \sqrt{d_1 + 3.6}}{d_1^3 \sqrt{d + 3.6}}. \tag{153}$$

From (150) and (153) we see that the values of N_1 are practically the same for either equation when the ratio of d to d_1 is small and that they differ widely for large ratios. For example, according to (150), 5.7 eight-inch pipes are equivalent in capacity to one sixteen-inch pipe, whereas (152) gives 6.15. The difference is negligible. Again, according to (150), 180 two-inch pipes are equivalent in capacity to one sixteen-inch pipe, whereas (153) gives 274. The difference is considerable. Equation (153) is most commonly accepted and is the basis of Table 87.

338. Friction through Valves and Fittings. — The formulas outlined in Table 83 are strictly applicable only to well-lagged pipes, free from bends or obstructions of any kind such as valves or fittings, which greatly increase the resistance of the flow of steam. If these obstructions must be considered, it is customary to allow for them by assuming an added length of straight pipe equivalent in resistance to the various fittings and bends. Unfortunately, the few tests which have been made for the purpose of determining the resistance of various pipe fittings give discordant results, and in the absence of more recent data the rules given by Briggs ("Warming Buildings by Steam") are probably as accurate as any.

According to Briggs, the length of pipe in inches equivalent to the resistance of one standard 90-degree elbow is

$$L = 76\, d \div \left(1 + \frac{3.6}{d}\right) \tag{154}$$

and to that of one globe valve

$$L = 114\, d \div \left(1 + \frac{3.6}{d}\right). \tag{155}$$

The resistance of gate valves is not considered.

TABLE 87.

TABLE OF EQUATION OF PIPES. — STANDARD STEAM AND GAS PIPES.

Dia.	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	Dia.
½	......	2.27	4.88	15.8	31.7	52.9	96.9	205	377	620	918	1,292	1,767	2,488	3,014	3,786	4,904	5,927	7,321	8,535	9,717	½
¾	2.60	......	2.05	6.97	14.0	23.3	42.5	90.4	166	273	405	569	779	1,096	1,328	1,668	2,161	2,615	3,226	3,761	4,282	¾
1	7.55	2.90	......	3.45	6.82	11.4	20.9	44.1	81.1	133	198	278	380	536	649	815	1,070	1,263	1,576	1,837	2,092	1
1½	24.2	9.30	3.20	......	2.00	3.34	6.13	13.0	23.8	39.2	58.1	81.7	112	157	190	239	310	375	463	539	614	1½
2	54.8	21.0	7.25	2.26	......	1.67	3.06	6.47	11.9	19.6	29.0	40.8	55.8	78.5	95.1	119	155	187	231	269	307	2
2½	102	39.4	13.6	4.23	1.87	......	1.83	3.87	7.12	11.7	17.4	24.4	33.4	47.0	56.9	71.5	92.6	112	138	161	184	2½
3	170	65.4	22.6	7.03	3.11	1.66	......	2.12	3.89	6.39	9.48	13.3	20.9	23.7	31.2	39.1	50.6	61.1	75.5	88.0	100	3
4	376	144	49.8	15.5	6.87	3.67	2.21	......	1.84	3.02	4.48	6.30	8.61	12.1	14.7	18.5	23.9	28.9	35.7	41.6	47.4	4
5	686	263	90.9	28.3	12.5	6.70	4.03	1.83	......	1.65	2.44	3.43	4.69	6.60	8.00	10.0	13.0	15.7	19.4	22.6	25.8	5
6	1,116	429	148	46.0	20.4	10.9	6.56	2.97	1.63	......	1.48	2.09	2.85	4.02	4.86	6.11	7.91	9.56	11.8	13.8	15.6	6
7	1,707	656	226	70.5	31.2	16.6	10.0	4.54	2.49	1.51	......	1.41	1.93	2.71	3.28	4.12	5.34	6.45	7.97	9.31	10.6	7
8	2,435	936	322	101	44.5	23.8	14.3	6.48	3.54	2.18	1.43	......	1.35	1.93	2.33	2.92	3.79	4.57	5.67	6.60	7.52	8
9	3,335	1,281	440	137	60.8	32.5	19.5	8.85	4.85	2.98	1.95	1.37	......	1.41	1.71	2.14	2.77	3.35	4.14	4.83	5.50	9
10	4,393	1,688	582	181	80.4	42.9	25.8	11.7	6.40	3.93	2.57	1.80	1.32	......	1.21	1.52	1.97	2.38	2.94	3.43	3.91	10
11	5,642	2,168	747	233	103	55.1	33.1	15.0	8.22	5.05	3.31	2.32	1.70	1.28	......	1.26	1.63	1.88	2.43	2.83	3.22	11
12	7,087	2,723	938	293	129	69.2	41.6	18.8	10.3	6.34	4.15	2.91	2.13	1.61	1.26	......	1.30	1.57	1.93	2.26	2.58	12
13	8,657	3,326	1,146	358	158	84.5	50.7	23.0	12.6	7.75	5.07	3.56	2.60	1.98	1.53	1.22	......	1.21	1.49	1.74	1.98	13
14	10,600	4,070	1,403	438	193	103	62.2	28.2	15.4	9.48	6.21	4.35	3.18	2.41	1.88	1.50	1.22	......	1.24	1.44	1.64	14
15	12,824	4,927	1,698	530	234	125	75.3	34.1	18.7	11.5	7.52	5.27	3.85	2.92	2.27	1.81	1.48	1.21	......	1.17	1.35	15
16	14,978	5,758	1,984	619	274	146	88.0	39.9	21.8	13.4	8.78	6.15	4.51	3.41	2.66	2.12	1.73	1.42	1.18	......	1.14	16
17	17,537	6,738	2,322	724	320	171	103	46.6	25.6	15.7	10.3	7.20	5.27	3.99	3.11	2.47	2.03	1.66	1.37	1.17	......	....
18	20,327	7,810	2,691	840	371	198	119	54.1	29.6	18.2	11.9	8.35	6.11	4.63	3.60	2.87	2.35	1.92	1.59	1.36	1.16	....
20	26,676	10,249	3,532	1,102	487	260	157	70.9	38.9	23.9	15.6	10.9	8.02	6.07	4.73	3.76	3.08	2.52	2.08	1.78	1.52	....
24	42,624	16,376	5,644	1,761	778	416	250	113	62.1	38.2	25.0	17.5	12.8	9.70	7.55	6.01	4.92	4.02	3.32	2.84	2.43	....
30	75,453	28,990	9,990	3,117	1,378	736	443	201	110	67.6	44.2	31.0	22.7	17.2	13.4	10.7	8.72	7.14	5.88	5.03	4.30	....
36	120,100	46,143	15,902	4,961	2,193	1,172	705	319	175	108	70.4	49.3	36.1	27.3	21.3	16.9	13.9	11.3	9.37	8.01	6.85	....
42	177,724	68,282	23,531	7,341	3,245	1,734	1,044	473	259	159	104	73.0	53.4	40.5	31.5	25.1	20.5	16.8	13.9	11.9	10.1	....
48	249,351	95,818	33,020	10,301	4,554	2,434	1,465	663	363	223	146	102	75.0	56.8	44.2	35.2	28.8	23.5	19.4	16.6	14.2	....
Dia.	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	

The upper portion above the diagonal line of blanks pertains to "standard" sizes while the lower portion is for pipe of the actual internal diameter given.

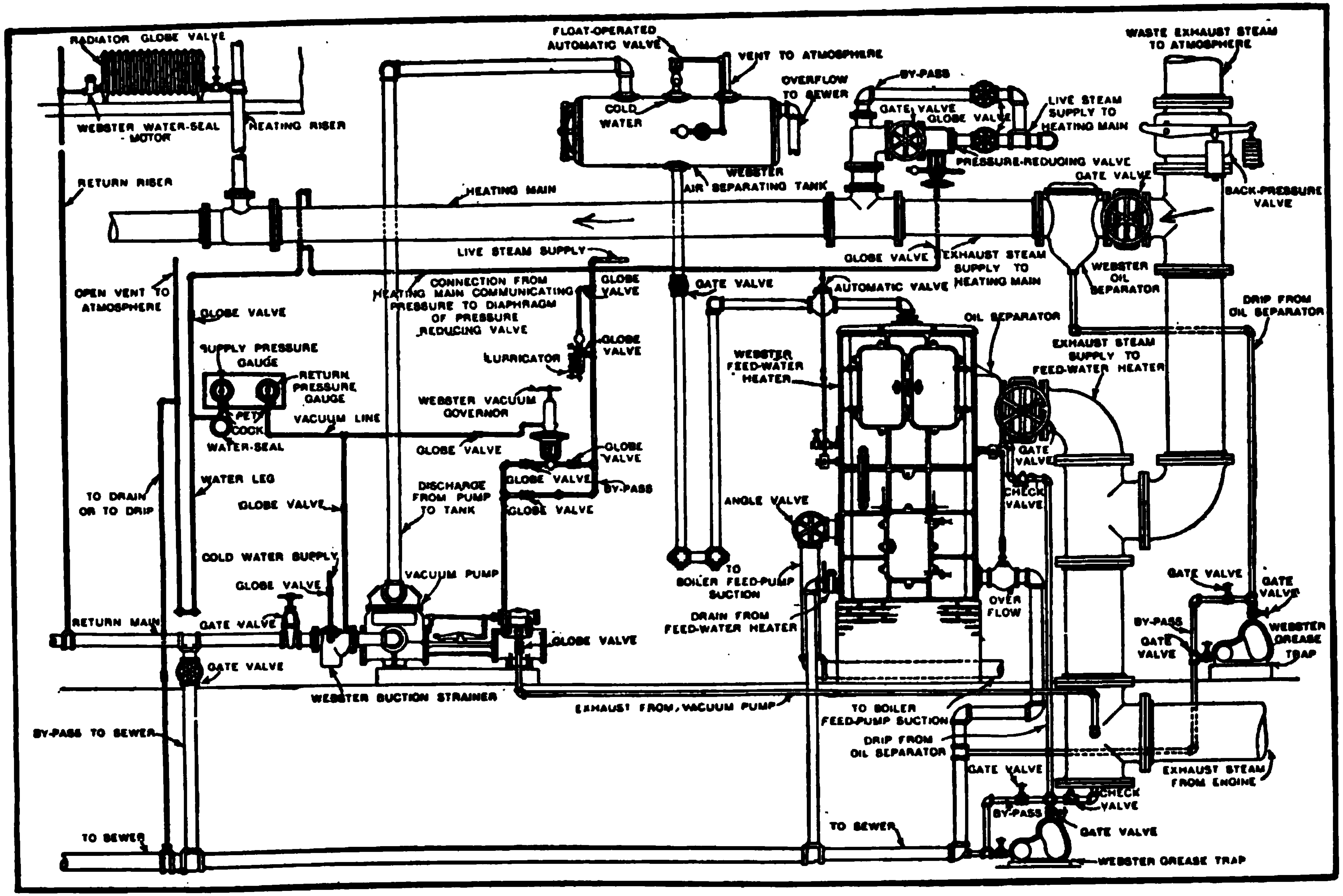


FIG. 363. Diagrammatic Arrangement of Webster Vacuum Heating System.

339. Exhaust Piping, Condensing Plants. — The exhaust piping in condensing plants is arranged either according to (1) the *independent* or (2) the *central condensing system.* In the former each engine is provided with an independent condenser and air pump. In case the vacuum "drops" or it is desired to operate non-condensing, the steam is discharged through a branch pipe with relief valve to the atmosphere, Figs. 3 and 219. When there are a number of engines in one installation the atmospheric pipes lead to a common *free exhaust main,* which, on account of its great size, is ordinarily constructed of light-weight riveted steel pipe. The short connection between engine and condenser is usually made with lap-welded steel pipe, since riveted joints are apt to leak, due to the engine vibrations. In a central condensing plant, Fig. 226, the several engines exhaust through a common main into a single large condenser. An atmospheric relief valve is usually provided in connection with the condenser, and no free exhaust main is necessary. Several arrangements of condenser piping are illustrated in Figs. 219 to 228.

340. Exhaust Piping, Non-Condensing Plant. — Webster Vacuum System. In the majority of non-condensing plants the exhaust steam is used for heating purposes. One of the best-known systems of exhaust steam heating, in which the back pressure on the engine is reduced by circulating below atmospheric pressure, is that known as the *Webster combination system.* The general arrangement is illustrated in Fig. 2 and the principles of operation are described in paragraph 3. It has the advantage of affording (1) minimum back pressure on the engine; (2) effective and continuous drainage of condensation from supply pipes and radiators; (3) continuous removal of air and entrained moisture from confined spaces; (4) independent regulation of temperature in each radiator; (5) continuous return of condensation to the boiler; (6) utilization of part of the exhaust steam for preheating the feed water; and (7) automatic regulation. Fig. 363 gives a diagrammatic arrangement of the piping and appurtenances in a typical installation. The characteristic feature of this system is the automatic outlet valve attached to each part requiring drainage, which permits both the water of condensation and the non-condensable gases to be removed continuously. The radiator temperature may be regulated by varying the quantity of steam supplied either by hand, or automatically by thermostatic control. The Webster valve, Fig. 364, enables the vacuum to withdraw the water of condensation as fast as it is formed irrespective of the pressure in the radiator, hence the supply may be throttled to such an extent that the temperature in the radiator is practically as low as that of steam cor-

responding to the pressure in the vacuum line. The small annular space between the inner tube of the float F and the guide H permits of a vacuum in the body of the valve. When the water from the radiator lifts the float the water is drawn into the returns pipe. The

OUTLET

FIG. 364. Webster Air Valve.

FIG. 365. Automatic Vacuum Valve, Illinois Engineering Co.

valve then returns to its seat and the escape of steam is prevented except such as finds its way through the annular space around the guide stem H. An improvement on this valve which prevents the escape of steam is illustrated in Fig. 365. When steam is admitted to the radiator the condensation flows into the valve, righting the float A and sealing the outlet B against the passage of steam; as the valve fills with water the buoyancy of the float raises it from its seat and permits the water to be drawn out; the float falls and reseats on the nipple when about a half-inch of water remains in the valve, thus maintaining a water seal.

Screen D prevents scale and dirt from entering the valve proper. By-pass H is for emergency use in draining off accumulated water in the radiator in case the valve becomes stopped up, and permits the bonnet to be removed without trouble from the accumulated water.

341. Exhaust Piping, Non-Condensing Plants. — Paul Heating System. The Paul vacuum system differs from the Webster in that the condensation, and the air and non-condensable gases are separately handled. Referring to Fig. 366, which gives a diagrammatic arrangement of the piping, the condensed steam gravitates to the *automatic return tank and pump* and is pumped either directly to the boiler or through the heater to the boiler. Air and vapor are withdrawn from the upper part of the radiator by the *Paul exhauster* or ejector, E, and discharged into the returns tank, which is vented to the atmosphere for the escape of the non-condensable gases. The exhauster

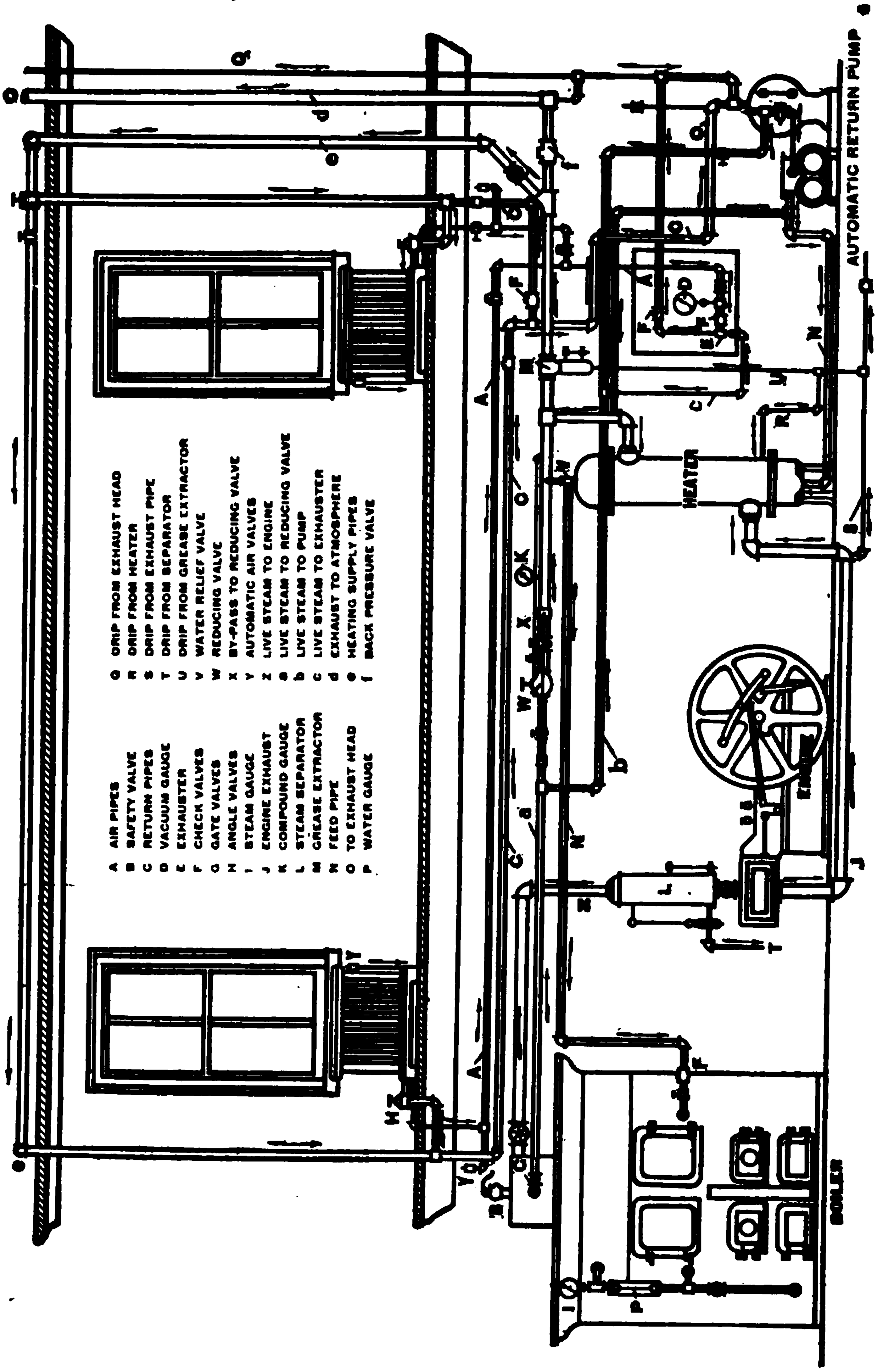


FIG. 366. Diagrammatic Arrangement of Piping in the Paul Vacuum Heating System.

receives its supply of steam through pipe *O*, Fig. 367, which shows the general arrangement of this apparatus. The piping is in duplicate to guard against failure to operate. The suction side of the exhauster is connected with the air pipes *A*, *A*, Fig. 366. Fig. 368 gives a section through the *Paul air* or *vacuum* valve which prevents steam from blowing into the air pipes and permits only air to pass. In Fig. 366 the heating system is piped on what is known as the "one-pipe down-feed system;" i.e., the exhaust steam is first conducted to a distributing header in the attic, from which the various supply pipes are led to the radiators. The water of condensation returns through these same pipes and gravitates to the returns pump. Both the supply steam and the condensation flow in the same direction. This system is also piped on the "one-pipe up-feed," the "two-pipe up-feed," and the "two-pipe down-feed" principle. The "one-pipe up-feed" differs from the system just described in that the steam flows upward through the risers and does away with the attic piping. The returns, however, flow against the current of steam and water hammer is more likely to occur than with the down-feed system. In the two-pipe systems the steam supply pipes or risers conduct steam only, and the returns carry the condensation. The one-pipe down-feed is cheaper and simpler and practically as efficient as the two-pipe system under normal conditions. It is objectionable, however, due to the difficulty of draining the radiator with

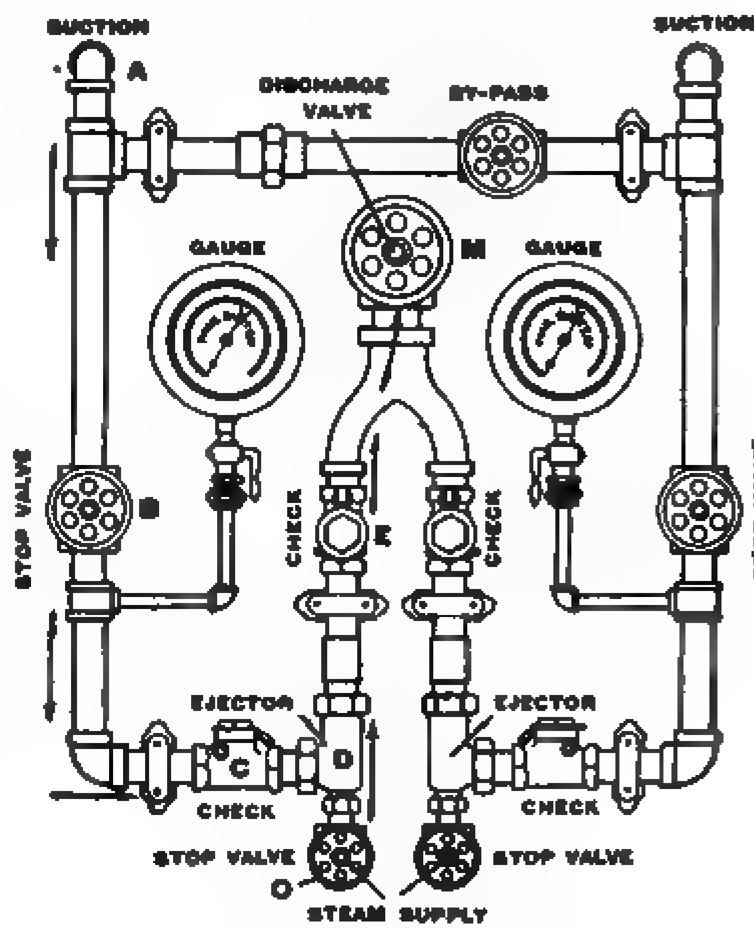


Fig. 367. Paul Exhauster.

COMPOSITION

CA

TO EXHAUSTER

Fig. 368. Paul Vacuum Valve.

closely throttled supply valve, since the velocity of the entering steam prevents the water from returning through the same orifice.

342. Automatic Temperature Control. — Experience shows that a considerable saving in fuel may be effected in the heating plants of tall office buildings and similar plants by automatically controlling the temperature. Hand-controlled valves are usually left wide open, and when the room becomes too hot the temperature is frequently lowered by opening the window, resulting in a waste of heat which may be considerable in modern buildings with hundreds of offices. Many successful methods of automatic temperature control are available, the usual system consisting of *thermostats* which control the supply of heat by means of *diaphragm valves*, the latter taking the place of the usual radiator supply valve.

Fig. 369 shows a Powers thermostat. The expansible disk *U* contains a volatile liquid having a boiling point of about 50 degrees F. The pressure of the vapor within the disk at a temperature of 70 degrees amounts to six pounds to the square inch, and varies with every change of temperature, causing a variation in the thickness of the disk. The disk is attached by a single screw *O* to the lever *Q*, which rests upon the screw *F* as a fulcrum. The flat spring *R* holds the lever and disk against the movable flange *M*. Connecting with the chamber *N* are two air passages *H* and *I*. The thermostat is attached by means of two screws at the upper end to a wall plate permanently secured to the wall. This wall plate has ports registering with *H* and *I*, one for supplying air under pressure and the other for conducting it to the diaphragm motor which operates the valve or damper. Air is admitted through *H* under a pressure of about fifteen pounds per square inch, and its passage into chamber *N* is regulated by the valve *J*, which is normally held to its seat by a coil spring under cap *P*. *K* is an elastic diaphragm carrying the flange *M*, with escape valve passage covered by the point of valve *L*. Valve *L* tends to remain open by reason of the spring. When the temperature rises sufficiently expansion of the disk *U* first causes the valve to seat, its

Fig. 369 Section Through Powers Thermostat.

spring being weaker than that above valve *J*. If the expansive motion is continued, valve *J* is lifted from its seat and compressed air flows into chamber *N*, exerting a pressure upon the elastic diaphragm *K* in opposition to the expansive force of the disk. If the temperature falls, the disk contracts and the overbalancing air pressure in *N* results in a reverse movement of the flange *M*, permitting the escape valve to open and discharge a portion of the air; thus the air pressure is maintained always in direct proportion to the expansive power (and temperature) of the disk *U*. The passage *I* communicates with a diaphragm valve, Fig. 370. The compressed air operates the diaphragm against a coiled spring resistance, so that the movement is proportional to the air pressure and the supply of steam controlled accordingly. The adjusting screw *G*, squared to receive a key, carries an indicator by means of which the thermostat can be set to carry any desired temperature within its range, usually from 60 to 80 degrees. In changing the temperature adjustment lever *Q* forces the disk *U* closer to or farther from the flange *M*.

FIG. 370. A Typical Diaphragm Valve.

In connecting up the system compressed air is carried to the thermostat and diaphragm valves, from a reservoir through small concealed pipes.

In the indirect system of heating the dampers are of the diaphragm type and the method of regulation is the same as with the direct system.

343. Feed-Water Piping. — The simplest arrangement of feed-water piping may be found in non-condensing plants, in which the feed water is obtained under a slight head, such as is afforded by the average city supply, and is heated in an open heater by the exhaust steam from the engine to a temperature varying from 180 to 210 degrees F. The hot feed water gravitates from the heater to the pump and then is forced to the boiler, or to the economizer if one is used. If a meter is used it is generally placed on the discharge side of the pump, and should be by-passed to permit it to be cut out for repairs. (Fig. 371.) Plants operating continuously should have feed pumps in duplicate. In some cases the returns from the heating system gravitate to the heater and only enough cold water is added to make up the loss from leakage, etc. In other cases the returns gravi-

tate to a special "returns tank," from which they are pumped directly to the boiler without further heating. Occasionally a live-steam purifier is used, especially if the water contains a large percentage of

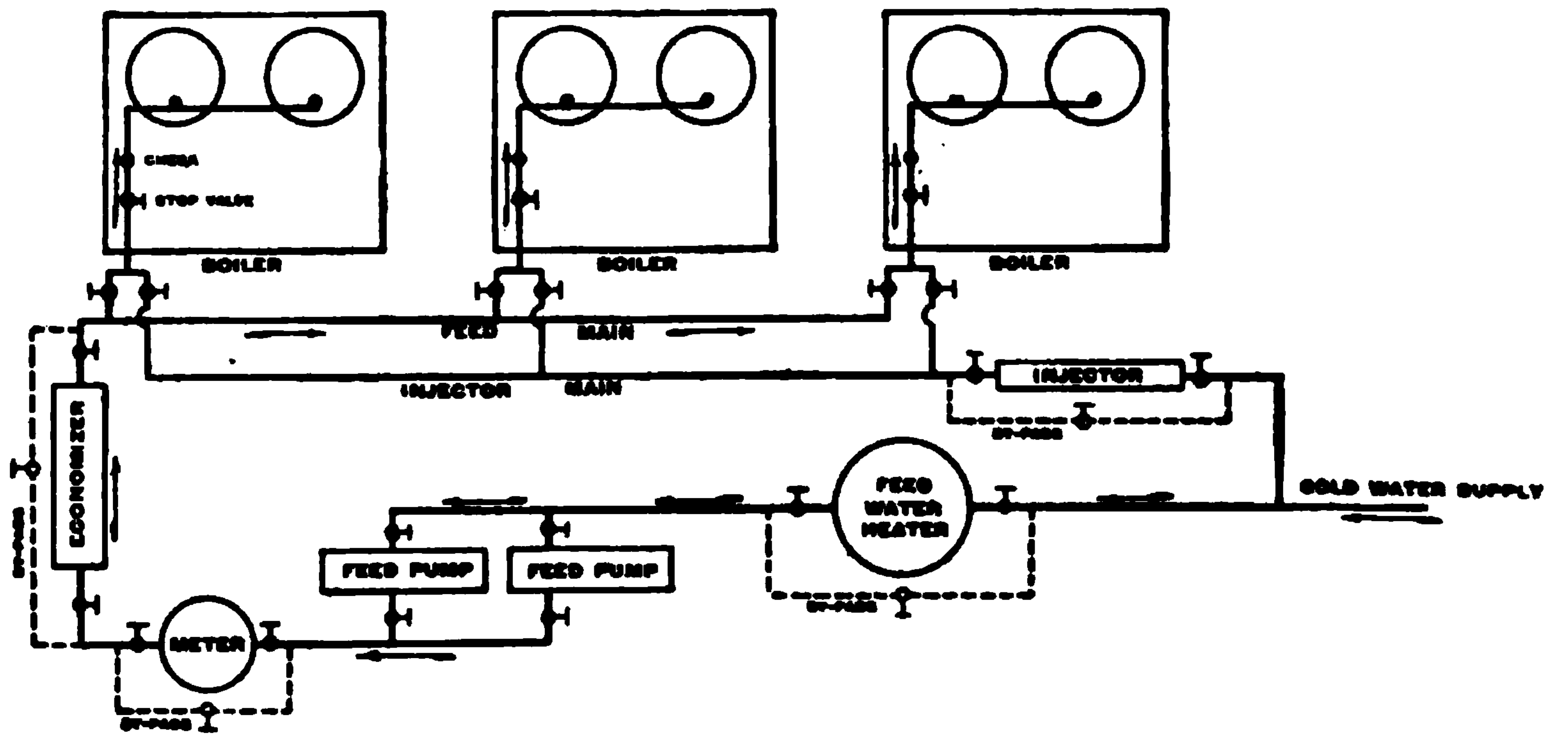


FIG. 371. Feed Water Piping; Non-Condensing Plant.

calcium sulphate. The feed is then subjected to boiler pressure and temperature and the greater part of the impurity precipitated before it enters the boiler. Closed heaters are often used in place of open heaters. When the supply is not under head a closed heater is usually preferred and is placed between the pump discharge and the feed main.

In condensing plants the feed piping is similar to that in non-condensing plants, except that if exhaust steam is used for heating

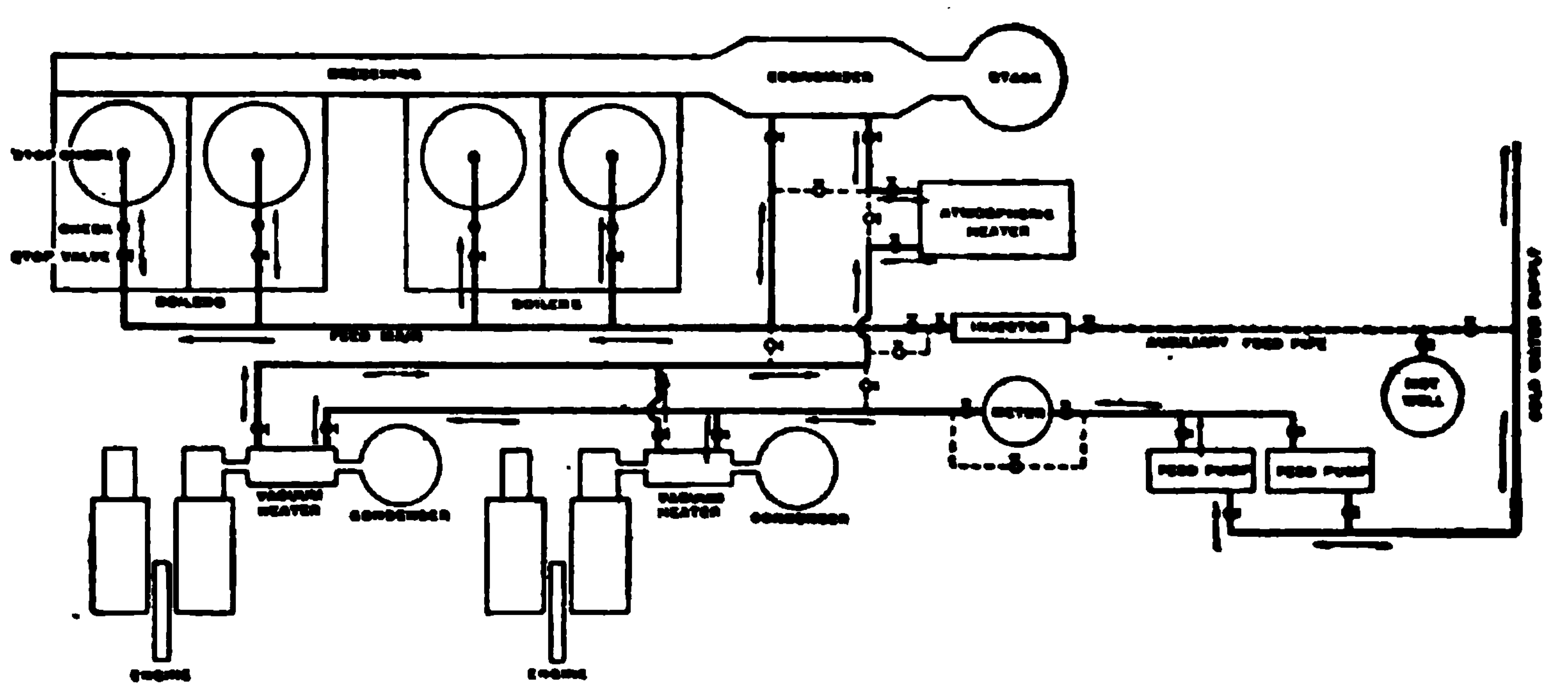


FIG. 372. Feed Water Piping; Condensing Plant.

purposes it is supplied by the auxiliaries, such as feed pumps, stoker engines, condenser engines, and other steam-using appliances.

In plants having a number of boilers it is customary to run a feed main or header the full length of the boiler room and connect it to

each boiler by a branch pipe. This main may be a simple header or in duplicate or of the "loop" or "ring" type. Horizontal tubular boilers are frequently arranged in one battery with the feed main run along the fronts of the boilers just above the fire doors. Water-tube boilers are generally set in a battery, and as the arrangement above would block the passageway between the batteries, the main is run either above or under the settings, the former being the more common. Where a single header is used, the feed pumps are sometimes placed so as to feed into opposite ends of the main, which is then cut into sections by valves. Another arrangement is to place the pumps so as to feed into the middle of the header. With the loop arrangement the main is ordinarily cut into sections by valves so that the water may be sent either way from the pumps and any defective section cut out. With duplicate mains a common arrangement is to place one main along the front of the boiler and the other at the rear or both overhead as in Fig. 359. Sometimes one main is placed in the passageway below the boiler setting and the other on top.

Standard wrought-iron pipe is usually used for pressures under 100 pounds and extra heavy pipe for greater pressures. The pipes and fittings from boiler to main are frequently of brass, and preferably so, since brass withstands corrosive action much better than iron or steel. Flanged joints should be used in all cases, since the pockets formed by the ordinary screwed joints hasten corrosion at those points. (*Power*, June, 1902, p. 4.)

Fig. 373, *A* to *E*, illustrates the various combinations of check valve, stop valves, and regulating valve in steam boiler practice. The

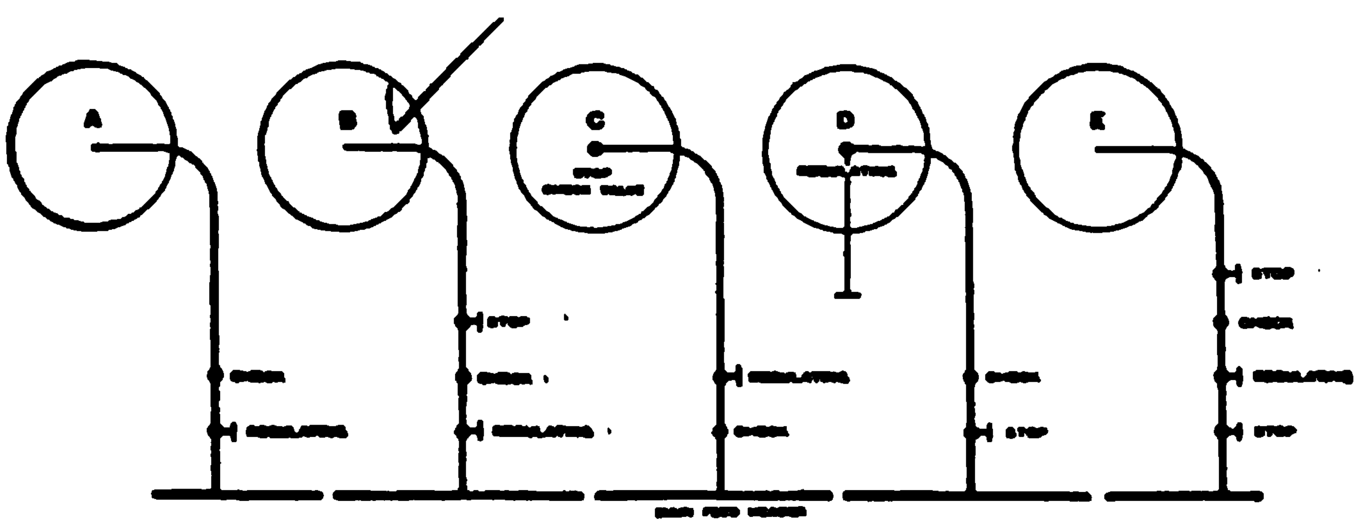


FIG. 373. Different Arrangements of Valves in Feed Water Branch Pipes.

simplest arrangement and one sometimes used in plants operating intermittently is shown in *A*. Here there are but two valves between the boiler and the main, the check being nearest the boiler and the stop valve at the main. The stop valve performs both the function of cutting out the boiler and of regulating the water supply. This

arrangement is not recommended, as any sticking or excessive leaking of the check valve will necessitate shutting down the boiler. *B* shows the most common arrangement. Here the check valve is placed between the regulating valve and a stop valve as indicated. This permits a disabled check to be easily removed while pressure is on the boiler and the main. *E* shows an arrangement whereby both check and regulating valve may be removed, and is particularly adapted to boilers operating continuously where the regulating valve is subjected to severe usage. In this case the stop valves are run wide open and are subjected to no wear. The regulating valve most highly recommended is a *self-packing* brass globe valve with *regrinding disk.* The check valve is ordinarily of the *swing check* pattern with regrinding disk, Fig. 384 (C). Modern practice recommends an *automatic water relief valve* in the discharge pipe immediately adjacent to each pump to prevent excessive pressure in case a valve is accidentally closed in by-passing or in changing over.

344. Flow of Water through Orifices, Nozzles, and Pipes. — Bernoulli's theorem is the rational basis of most empirical formulas for the steady flow of a fluid from an up-stream position *n* to a down-stream position *m*, thus ("Mechanics of Engineering," Church, p. 706):

$$\frac{P_m}{\gamma} + \frac{V_m^2}{2g} + Z_m = \frac{P_n}{\gamma} + \frac{V_n^2}{2g} + Z_n - \left\{\begin{matrix}\text{all losses of head} \\ \text{occurring between} \\ n \text{ and } m\end{matrix}\right\}, \tag{156}$$

in which

V = velocity in feet per second at the point considered.
P = pressure in pounds per square foot.
Z = potential head in feet of the fluid.
γ = density of the fluid, pounds per cubic foot.
g = acceleration of gravity.

Each loss of head will be of the form $K\frac{V^2}{2g}$, in which K is the *coefficient of resistance* to be determined experimentally. The loss of head due to *skin friction* is expressed:

$$H = 4f\frac{l}{d} \times \frac{v^2}{2g}, \tag{157}$$

in which

f = the coefficient of friction of the fluid in the pipe.
l = length of the pipe in feet.
d = diameter of the pipe in feet.
Other notations as in (156).

Discharge from a circular vertical orifice with sharp corners:

$$Q = CA\sqrt{2\,gh}, \tag{158}$$

in which

Q = cubic feet per second.
C = coefficient, varying from 0.59 to 0.65 (Merriman, Treatise on Hydraulics, p. 118).
A = area of the orifice, square feet.
h = head of water in feet.
g = acceleration of gravity = 32.2.

Discharge from short cylindrical nozzles three diameters in length, with rounded entrance ("Mechanics of Engineering," Church, p. 690):

$$Q = 0.815\,A\sqrt{2\,gh}. \tag{159}$$

Discharge from short nozzles with well-rounded corners and conical convergent tubes, angle of convergence 13½ degrees (Church, p. 693):

$$Q = 0.94\,A\sqrt{2\,gh}. \tag{160}$$

Discharge from cylindrical pipe under 500 diameters in length (Church, p. 712):

$$Q = 6.3\sqrt{\frac{d^5h}{(1+.5)\,d+4\,fl}}, \tag{161}$$

in which

f = coefficient of friction.
Other notations as above.

f varies with the nature of the inside surface, the diameter of the pipe, and the velocity of flow.

Discharge through very long cylindrical pipes ("Mechanics of Engineering," Church, p. 715):

$$Q = 3.15\sqrt{\frac{d^5h}{fl}}. \tag{162}$$

Loss of head due to friction in water pipes. Weisbach's formula is as follows:

$$H = \left(0.0144 + \frac{0.01716}{\sqrt{V}}\right)\frac{LV^2}{5.367\,d}, \tag{163}$$

in which

H = friction head in feet.
V = velocity in feet per second.
L = length of pipe in feet.
d = diameter of pipe in inches.

TABLE OF THE COEFFICIENT f FOR FRICTION OF WATER IN *CLEAN* IRON PIPES.

(Abridged from Fanning.)

Velocity in Ft. per Sec.	Diam. = ½ in. = .0417 ft.	Diam. = 1 in. = .0834 ft.	Diam. = 2 in. = .1667 ft.	Diam. = 3 in. = .25 ft.	Diam. = 4 in. = .333 ft.	Diam. = 6 in. = .50 ft.	Diam. = 8 in. = .667 ft.
0.1	.0150	.0119	.00870	.00800	.00763	.00730	.00704
0.3	.0137	.0113	850	784	750	720	693
0.6	.0124	.0104	822	767	732	702	677
1.0	.0110	.00950	790	743	712	684	659
1.5	.00959	.00868	.00757	.00720	.00693	.00662	.00640
2.0	.00862	810	731	700	678	648	624
2.5	795	768	710	683	662	634	611
3.0	.00753	.00734	.00692	.00670	.00650	.00623	.00600
4.0	722	702	671	651	631	607	586
6.0	689	670	640	622	605	582	562
8.0	663	646	618	600	587	562	544
12.0	630	614	590	582	560	540	522
16.0	.00618	.00600	.00581	.00570	.00552	.00530	.00513
20.0	615	598	579	566	549	525	508

Velocity in Ft. per Sec.	Diam. = 10 in. = .833 ft.	Diam. = 12 in. = 1.00 ft.	Diam. = 16 in. = 1.333 ft.	Diam. = 20 in. = 1.667 ft.	Diam. = 30 in. = 2.50 ft.	Diam. = 40 in. = 3.333 ft.	Diam. = 60 in. = 5. ft.
0.1	.00684	.00669	.00623	....	....	....	....
0.3	673	657	614	.00578	....	....	....
0.6	659	642	603	567	.00504	.00434	.00357
1.0	643	624	588	555	492	428	353
1.5	.00625	.00607	.00572	.00542	.00482	.00421	.00349
2.0	609	593	559	529	470	416	346
2.5	596	581	548	518	460	410	342
3.0	.00584	.00570	.00538	.00509	.00452	.00407	.00339
4.0	568	553	524	498	441	400	333
6.0	548	534	507	482	430	391	324
8.0	532	520	491	470	422	384	320
12.0	512	500	478	457	412	377	.00313
16.0	.00502	.00491	.00470	.00450	.00406	.00370	....
20.0	498	485	....	....	....	....	....

William Cox (*American Machinist*, Dec. 28, 1893) gives a simple formula which gives almost identical results:

$$H = \frac{(4\,V^2 + 5\,V - 2)\,L}{1200\,d}. \tag{164}$$

Notations as in (163).

Loss of head due to friction of fittings. Formulas (161) to (164) are based on the flow of water through clean straight cylindrical pipes.

Where there are bends, valves, or fittings in the line the flow is decreased on account of the additional resistance.

These frictional losses are conveniently expressed in feet of water, thus:

$$H = C\frac{V^2}{2g}. \qquad (165)$$

C having the following values:

	Angles.		Class of Valve.		
	45 degrees.	90 degrees.	Gate.	Globe.	Angle.
C	0.182	0.98	0.182	1.91	2.94

Example: Determine the pressure necessary to deliver 200 gallons of water per minute through a 4-inch iron pipe line 400 feet long, fitted with four right-angle elbows and two globe valves. The water is to be discharged into an open tank.

A flow of 200 gallons per minute gives a velocity of

$\frac{200 \times 144}{7.48 \times 60 \times 12.72} = 5$ feet per second (7.48 = number of gallons per cubic foot, and 12.72 = internal area of the pipe, square inches).

From the preceding table, $f = 0.618$ for $V = 5$.

From (165),

Resistance head of 4 elbows $= 0.98 \times \frac{25}{64.4} \times 4 = 1.52$ feet.

Resistance head of 2 globe valves:

$$1.91 \times \frac{25}{64.4} \times 2 = 1.48 \text{ feet.}$$

Resistance head of all fittings:

$$1.52 + 1.48 = 3 \text{ feet.}$$

Substitute $V = 5$, $L = 400$, and $d = 4$ in (164).

$$H = \left(\frac{4 \times 5^2 + 5 \times 5 - 2}{1200 \times 4}\right) 400$$
$$= 10.25 \text{ feet, resistance head of the pipe.}$$

Total resistance head = 10.25 + 3 = 13.25 feet of water, or 5.75 pounds per square inch.

Example: How many gallons of water will be discharged per minute through above line with initial pressure of 100 pounds per square inch, and what will be the pressure at the discharge end?

Since f depends upon the unknown V, we may put $f = 0.006$ for a first approximation and solve for V; then take a new value of f and substitute again, and so on.

Substitute $f = 0.006$, $d = \frac{4}{12}$, $h = 100 \times 2.3 = 230$, and $l = 400$ in (162):

$$Q = 3.15 \sqrt{\frac{0.33^5 + 230}{0.006 \times 400}}$$

= 1.95 cubic feet per second, corresponding to a velocity of 22 feet per second.

From the preceding table,

$f = 0.00548$ (by interpolation) for $V = 22$ feet per second.

From (165) the friction of 4 elbows and 2 globe valves is found to be 58 feet for $V = 22$.

From (164) a resistance head of 58 feet of water for $V = 22$ is found to be equivalent to 136 feet of straight pipe, thus:

$$58 = \left(\frac{4 \times 22^2 \times 5 \times 22 - 2}{1200 \times 4}\right) L.$$

$$L = 136.$$

Substitute $f = 0.0548$, $l = 400 + 136 = 536$ in (162):

$$Q = 3.15 \sqrt{\frac{0.33^5 \times 230}{0.0058 \times 536}}$$

= 1.74 cubic feet per second, corresponding to a velocity of 19.3 feet per second.

= 780 gallons per minute.

If greater accuracy is necessary determine f and L for $V = 19.3$ and proceed as above.

The total friction head may be determined from (164) thus:

$$H = \left(\frac{4 \times 19.3^2 + 5 \times 19.3 - 2}{1200 \times 4}\right) 536$$

= 177 feet of water.

= 77 pounds per square inch.

The pressure at the discharge end will be

$100 - 77 = 23$ pounds per square inch.

Average power plant practice gives the following maximum velocities of flow in water pipes:

Size of Pipe in Inches.	Velocity, Feet per Minute.	Size of Pipe in Inches.	Velocity, Feet per Minute.
$\frac{1}{8}$ to $\frac{1}{2}$	50	3 to 6	250
$\frac{1}{2}$ to $1\frac{1}{2}$	100	Over 6	300–400
$1\frac{1}{2}$ to 3	200		

345. Stop Valves. — The valves used to control and regulate the flow of fluids are the most important element in any piping system. A good valve should have sufficient weight of metal to prevent distortion under varying temperature and pressure, or under strains due to connection with the piping; the seats should be easily repaired or renewed; there should be no pockets or projections for the accumulation of dirt and scale, and the valve stem should permit of easy and efficient packing. Stop valves are made in such a variety of designs that a brief description will be given of only a few of the best-known types.

Fig. 374 shows a section of an ordinary *globe valve*, so called because of the globular form of the casing. This type of valve is the most

FIG. 374. A Typical Globe Valve, Screw-Top, inside Screw.

FIG. 375. A Typical Globe Valve, Bolt-Top, outside Screw.

common in use. Globe valves are designated as (1) *inside screw* and (2) *outside screw*, according as the screw portion of the stem is inside the casting, Fig. 374, or outside, Fig. 375. The top or bonnet may be screwed into the body of the valve, Fig. 374, or bolted, Fig. 375. The smaller sizes, three inches and under, are usually of the *screw-top* type and the larger of the *bolt-top* type. Valves with *outside yoke* and *screw* are preferable to the other in that they show at a glance whether the valve is open or closed, an advantage in changing from one section to another. The disks are made in a variety of forms, the material

depending upon the nature of the fluid to be controlled. Thus, for cold water, hard rubber composition gives good results; for hot water and low-pressure steam, Babbitt metal; for high-pressure steam, copper or bronze; and for highly superheated steam, nickel. The valve bodies are of brass for sizes under three inches, cast iron for the larger sizes and ordinary pressures and temperatures, and cast steel or semi-steel for high temperatures and pressures. Globe valves should always be set to close against the pressure, otherwise they could not be opened if the valves should become detached from the stem. Globe valves should never be placed in a horizontal steam return pipe with

FIG. 376. A Typical Gate Valve, Solid-Wedge, Screw-Top, outside Screw.

FIG. 377. A Typical Gate Valve, Solid-Wedge, Bolt-Top, inside Screw.

the stem vertical, because the condensation will fill the pipe about half full before it can flow through the valve. Globe valves that are open all the time are preferably designed with a *self-packing spindle*, as in Fig. 375, in which the top of shoulder C can be drawn tightly against the under surface of bonnet S, thus preventing steam from leaking past the screw threads while the spindle is being packed.

Figs. 376 to 379 show different types of *gate* or *straight* way valves. These valves offer little resistance to the flow of steam or liquid passing through them, and are generally used in the best class of work. Fig. 376 shows a section through a *solid-wedge* gate valve with outside

screw and yoke. This form of outside screw and yoke with stem protruding beyond the hand wheel is a perfect indicator to show whether the valve is open or shut, as the hand wheel is stationary and the spindle rises in direct proportion to the amount the valve is opened. For these reasons outside screw valves are preferable for high-pressure work and especially for the larger sizes. The seats are made solid, or removable, and of various materials for different pressures and temperatures. Fig. 378 shows a section through a *split-wedge* gate valve with parallel faces and seats. For the sake of illus-

Fig. 378. A Typical Gate Valve, Split-Wedge, Bolt-Top, inside Screw.

Fig. 379. Ludlow Angle Valve, Gate Pattern.

tration this valve is fitted with inside screw. In this design the spindle remains stationary so far as any vertical movement is concerned, and the gate or plug being attached to it by means of a threaded nut rises into the bonnet when the spindle is revolved. It is impossible to tell by its appearance whether this form of valve is open or closed. Valves with inside screw are adapted to situations where there is considerable dirt and grit, since the screw is inclosed and protected and excessive wear is thus avoided. Gate valves with split gates are more flexible than those with solid gates, and hence are less likely to leak. Fig. 379 shows the application

of the gate system to an angle valve. All high-pressure valves above 8 inches in diameter should be provided with a small by-pass valve, as the pressure exerted against the disk or gate is very great when the valve is closed and the force required to move it is considerable. The by-pass valve also facilitates "warming up" the section to be cut in and is more readily operated than the main valve.

346. Automatic Non-Return Valves. — Fig. 380 shows a section through an *automatic non-return* valve as applied to the nozzle of a steam boiler. As will be seen from the illustration it amounts to practically a large check valve with cushioned disk. The object of this device is the equalization of pressure between the different units of the battery, the valve remaining closed as long as the individual boiler pressure is lower than that of the header. In case a tube blows out the valve closes automatically, owing to the reduction of pressure and prevents the header steam from entering the boiler. It acts also as a safety stop to prevent steam being turned into a cold boiler while men are working inside, because it cannot be opened when there is pressure on the header side only. To be successful, such a valve should not open until the pressure in the boiler is equal to that in the header; it should not stick and become inoperative nor chatter and hammer while performing its work. Referring to Fig. 380, tail rod E insures alignment and hence prevents sticking; steam space C acts as a dashpot to prevent hammering of the valve as it rises, and steam space D acts as a cushion and prevents hammering at closing. Lip F is made to enter the opening in the seat and reduce wire drawing across the seat. Fig. 358 shows the installation of a number of non-return valves at the Yonkers power house of the New York Central Railway Company.

FIG. 380. Anderson Non-Return Valve.

347. Emergency Valves. — In large power plants it is customary to protect the various divisions of the steam piping by *emergency valves* which may be closed by suitable means at any reasonable distance from the valve. The simplest form of emergency stop is a weighted "butterfly" valve, which is to all intents and purposes a weighted check as illustrated in Fig. 385 (F). The weight when supported, say

by a cord and pulley, holds the valve open; when the cord is cut or released the weight drops and forces the valve shut. The cord may lead to any convenient and safe distance from the valve. In applying this system of control to steam engines the valve is placed in the steam pipe just above the throttle and the weight held up by a lever controlled by the main governor or preferably by a separate governor. Should the engine exceed a certain speed, as in case of accident to the regular governor, the lever supporting the weight is tripped by the emergency governor and the valve is closed automatically. For high pressures a rotating plug valve or cock is preferred to the butterfly type, since it is balanced in all positions. Gate and globe valves may be converted into emergency valves by having the stems mechanically operated by electric motors, hydraulic pistons, and the like. Fig. 381 shows a section through a Crane hydraulically operated emergency gate valve.

FIG. 381. Crane Emergency Valve, Hydraulic.

FIG. 382. Anderson Triple-Duty Emergency Valve.

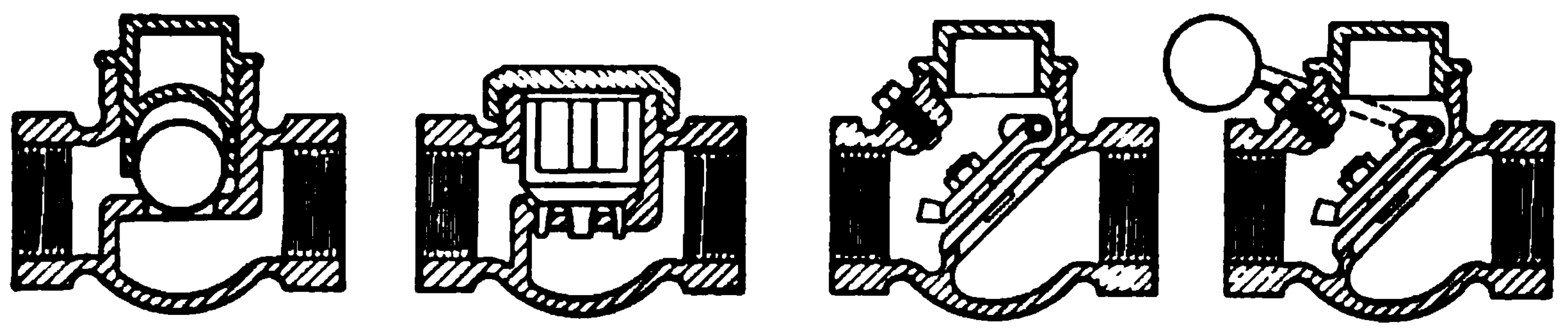
FIG. 383. Pilot Valve for Anderson Triple-Duty Emergency Valve.

Fig. 382 shows a partial section through an "Anderson triple-duty" emergency valve, and Fig. 383 a section through the pilot valve. A steam connection from the main line to the top of a copper diaphragm holds the pilot valve closed because of the large area above the diaphragm. A steam pipe connection from underneath the emergency piston of the triple-acting valve also leads to the pilot

valve. In case a break occurs in the main steam line or branches, the pressure is removed from the top of the pilot valve, causing it to open, thus exhausting the pressure from beneath the emergency piston in the triple-acting valve. The boiler pressure on top of the emergency piston causes the valve to close. Pilot valves may be located at any desirable places, thus affording control from different points.

In the "Locke automatic engine stop system" the stop valve is operated by an electric motor which is controlled by contact points operated by a speed-limit device. (See *Power*, August, 1907, p. 471, for a detailed description.)

348. Check Valves. — Fig. 384, *A* to *D*, illustrate the different types of check valves in most common use. *A* is a *ball check*, *B* a *cup*



(A) (B) (C) (D)

Fig. 384. Types of Check Valves.

or *disk check*, *C* a *swing check*, and *D* a *weighted check*. Occasionally the valve body is fitted with a valve stem and handle for holding the disk against its seat, in which it is designated as a *stop check*. In *A* and *B* the valve seat is parallel to the direction of flow and the valve is held in place by its own weight and by the pressure of the fluid in case of reverse flow. In the swing check the seat is at an angle of about 45 degrees to the direction of flow. The latter construction is preferred as it offers less resistance to flow and there is less tendency for impurities to lodge on the valve seat. By extending the hinge of the swing through the body of the valve, a lever and weight may be attached as in *D* and the check will not open except at a pressure corresponding to the resistance of the weight. It thus acts as a relief valve and at the same time prevents a reversal of flow. *Stop checks* are usually inserted in boiler feed lines close to the boiler, and when locked, act as any ordinary stop valve and permit the piping to be dismantled or the regulating valve to be reground without lowering the pressure on the boiler. Since the wear on check valves is excessive and necessitates frequent regrinding they are often mounted with *regrinding disks*, Fig. 384 (*C*), which may be "ground" against the seat without removing the valve from the line.

349. Blow-off Cocks and Valves. — The requirements of a good blow-off valve are that it shall furnish a free passage for scale and sediment, that it shall close tightly so as not to leak, and that it shall open easily without sticking or cutting. On account of the rather severe service to which such valves are subjected, they are made very heavy, with renewable wearing parts.

Fig. 385 gives a sectional view of a Crane ferrosteel valve. The bonnet is easily taken off and the disk removed to be refaced or replaced by a new one. The old disk is repaired by pouring in a hard Babbitt metal and facing it off flush. The seats are of brass and oval on top to prevent scale lodging between them and the disk, and are so made that they may be removed; but it has been found in practice that there is not much cutting of the seat, the damage usually being confined to the softer Babbitt metal which faces the disk.

FIG. 385. Crane Ferrosteel Blow-off Valve.

FIG. 387. A Typical Blow-off Cock.

FIG. 386. Faber Blow-off Valve.

Fig. 386 gives a sectional view of a Faber valve. When the disk, which makes a snug fit in the body of the valve, is in the position shown, the boiler discharge is practically shut off and any sediment lying on the seat is cleaned off by a jet of steam or water.

Fig. 387 shows a section through a typical *blow-off cock* of the straightway taper plug pattern with self-locking cam. Plug cocks are often used instead of valves on the blow-off piping.

Current practice recommends the use of two valves, or rather one valve and one cock, in the blow-off line of each boiler. In most of

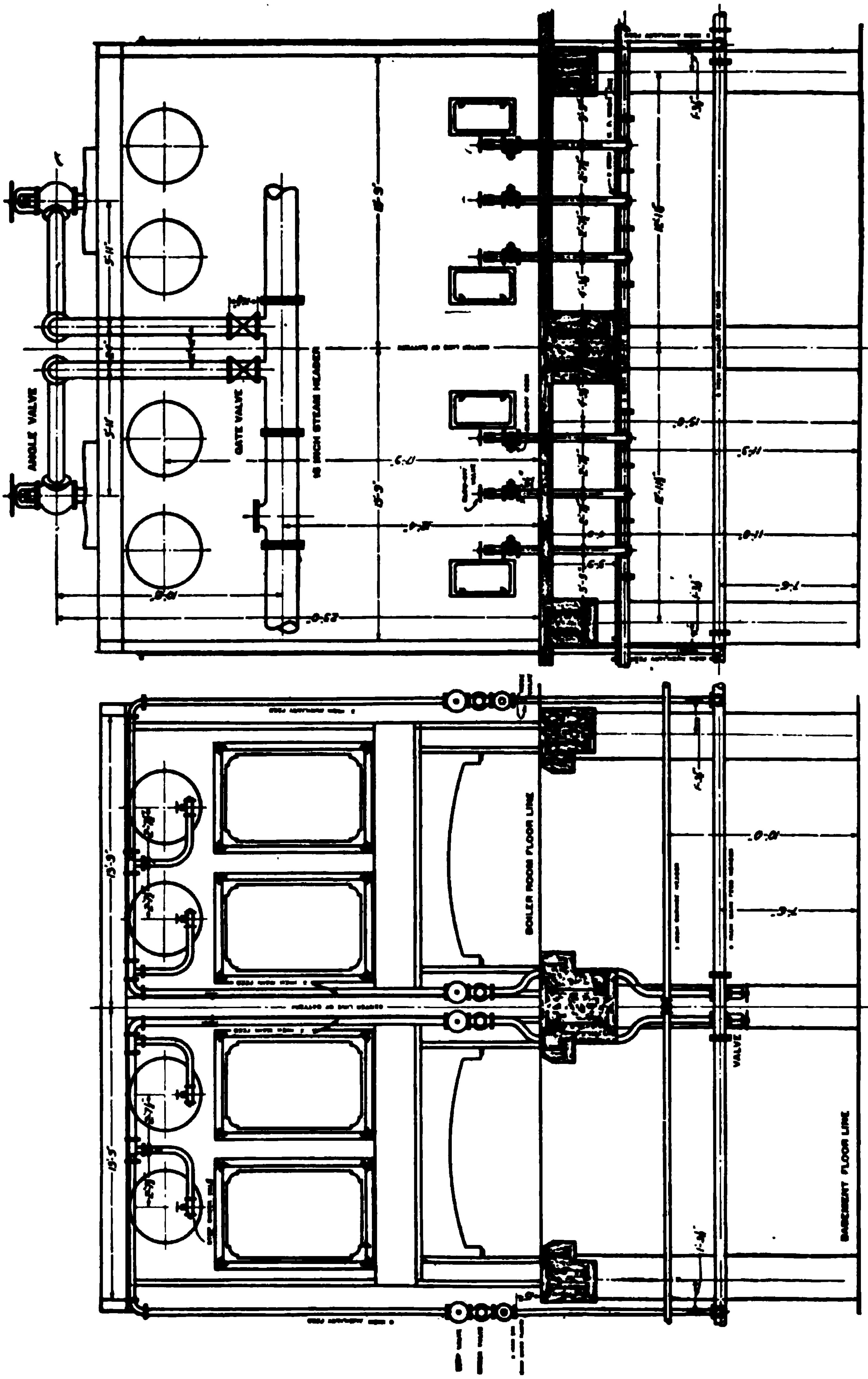


FIG. 388. Feed Water and Blow-off Piping, South Side Elevated Ry. Station, Chicago, Ill.

the large stations a blow-off valve and a blow-off cock are installed as indicated in Fig. 388. The number and size of blow-off cocks are usually specified by city or state legislation.

350. Safety Valves. — Fig. 389 shows a section through the simplest form of safety valve. The valve is held on its seat against the boiler pressure by a cast-iron weight as indicated. This type has the advantage of great simplicity, and can be least affected by tampering, since it requires so much weight that any additional amount which would seriously overload it can be quickly detected. For high pressure and large sizes of boiler this class of valve is entirely too cumbersome.

Fig. 389. "Dead-weight" Safety Valve.

Fig. 390 shows the general details of the common *lever safety valve.* The valve is held against its seat by a loaded lever, thereby enabling the use of a much smaller weight than the "dead weight" type, since the resistance is multiplied by the ratio of the long arm of the lever to

Fig. 390. Common Lever Safety Valve.

the short one. The proper position of the weight is determined by simple proportion.

Fig. 391 shows a section through a typical *pop safety valve* in which the boiler pressure is resisted by a spring. This type of valve has practically supplanted all other forms. The boiler pressure acting upon the under side of valve V is resisted by the tension in spring S. As soon as the boiler pressure exceeds the resistance of the spring the valve lifts from its seat and the steam escapes through opening O. The static pressure of the steam plus the force of its reaction in being deflected from the surface A holds the valve open until the pressure in

the boiler drops about 5 pounds below that at which the valve is lifted. The additional area of valve exposed to pressure when the valve lifts causes it to open with a sudden motion which has given it its name, and it also closes suddenly when the pressure has fallen. These valves are arranged so that the spring tension may be varied without taking them apart, and provision is made for lifting the seats by means of a lever. The seats are of solid nickel in the best designs, to minimize corrosion.

Fig. 391. Consolidated Pop Safety Valve.

The commercial rating of a safety valve is based upon the area exposed to pressure when the valve is closed.

The number and size of safety valves for a given boiler are ordinarily specified by city or state legislation.

The logical method for determining the size of safety valves is to make the actual opening at discharge sufficient to take care of all steam generated at maximum load. Most rules, however, are empirical and based on the extent of grate surface, thus:

According to the Boiler Inspection Department of the city of Philadelphia,

$$A = \frac{22.5\,G}{P + 8.6}, \qquad (166)$$

in which

A = area of combined safety valves, inches.
G = Grate area, square feet.
P = boiler pressure, pounds per square inch gauge.

According to the rule of United States Supervising Inspectors of Steam Vessels,*

$$A = \frac{G}{2}, \text{ for lever safety valves.} \qquad (167)$$

$$A = \frac{G}{3}, \text{ for pop safety valves.} \qquad (169)$$

Other notations as in (166).

Hutton's rule is ("Steam-Boiler Construction,") p. 470:

$$A = \frac{4\,G}{\sqrt{P}}. \qquad (169)$$

All notations as above.

* Superseded 1908 by the following:

$$A = 0.2074\,\frac{W}{P}$$

A as above
W = weight of steam per hour, lbs.
P = Absolute steam pressure.
See *Power*, Mar. 9, 1908, p. 480; Mar. 16, 1909, p. 520.

The Consolidated Safety Valve Company's circular gives the following rated capacity of its nickel-seat pop safety valves:

Sizes in Inches.	1	1¼	1½	2	2½	3	3½	4	4½	5	5½	6
Boiler Horse-power —												
From	8	10	20	[illegible]	[illegible]	75	100	125	150	175	200	275
To	10	15	[illegible]	50	75	100	125	150	175	200	275	300

351. Back-Pressure and Atmospheric Relief Valves. — These valves are for the purpose of preventing excessive back pressure in exhaust pipes. In non-condensing plants such valves are designated as *back-pressure valves* and in condensing plants as *atmospheric relief valves*. In the former the valve is usually adjusted so that a pressure of one to

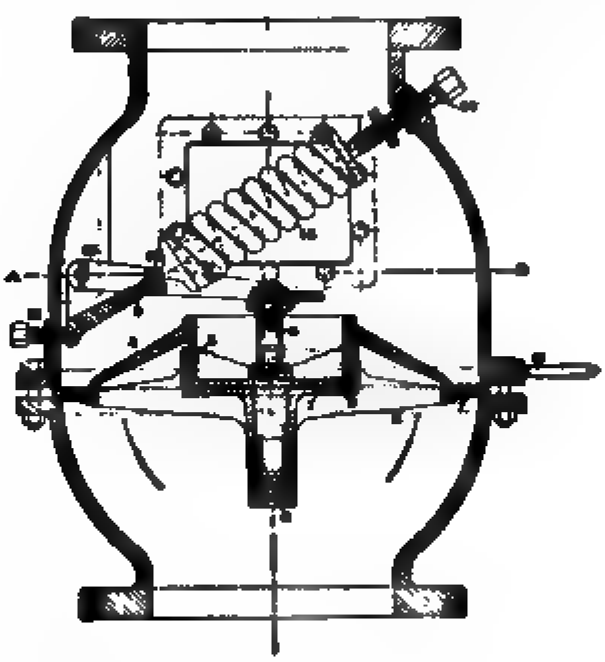


Fig. 392. Foster Back-Pressure Valve.

Fig. 393. Davis Back-Pressure Valve.

five pounds above the atmosphere is necessary to lift it from its seat; in the latter the valve lifts at about atmospheric pressure. They are practically identical in construction, differing only in minor details. A slight leakage in the back-pressure valve is of small consequence, but in an atmospheric relief valve it may seriously affect the degree of vacuum and throw unnecessary work upon the air pump, hence it is customary to "water-seal" the latter. Fig. 392 shows a section through a typical back-pressure valve. The valve proper consists of a single disk moving vertically. The valve stem is in the form of a piston or dashpot which prevents sudden closing or hammering. The pressure holding the valve against its seat is regulated by a spring. When the back pressure becomes greater than atmospheric plus that added by the spring, the valve raises from its seat and relieves it.

Fig. 393 shows a section through a Davis back-pressure valve, in which the resisting pressure is varied by means of a lever and weight.

Fig. 363 shows the application of a back-pressure valve to a typical heating system.

Fig. 394 shows a section through a typical *atmospheric relief valve.* Opening B is connected to the exhaust pipe and opening A leads to the atmosphere. Under normal conditions of operation atmospheric pressure holds valve V against its seat. Water in groove S " water-seals " the seat and prevents air from being drawn into the condenser. In case the pressure in pipe B becomes greater than atmospheric it lifts valve V from its seat and is relieved. Piston P acts as a dash-pot and prevents the valve from slamming.

Fig. 395 shows a section through an atmospheric relief valve in which the weight of the valve is counterbalanced or even over-balanced by an adjustable weight and lever, thereby permitting the valve to open at or below atmospheric pressure, as may be desired.

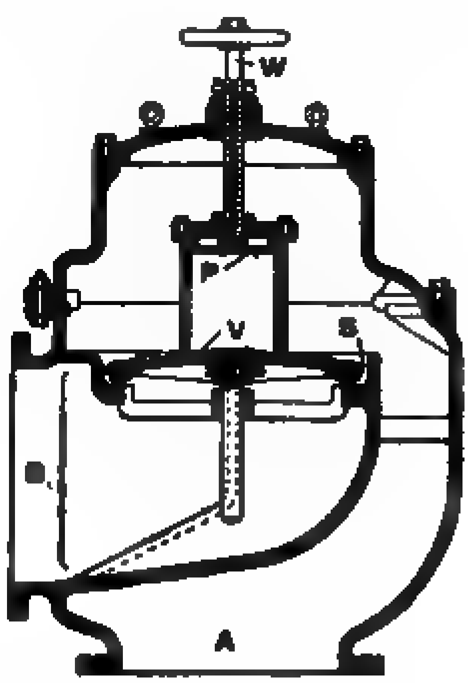


FIG. 394. Crane Atmospheric Relief Valve. FIG. 395. Acton Atmospheric Relief Valve.

352. Reducing Valves. — It is often necessary to provide steam at different pressures in the same plant, as in the case of a combined power and heating plant. To effect this result the reduction in pressure is accomplished by passing the steam through a *reducing valve,* which is but an automatically operated throttle valve. There are many different forms, the operation of all being based upon the same general principles.

In the Kieley valve, Fig. 396, the low-pressure steam acts upon the top of flexible diaphragm D, and the weighted lever L (which may be adjusted to give the desired reduction in pressure) acts upon the other side. The movement of the diaphragm causes the balanced valve V

at the upper end of the spindle to open or close, as may be necessary to maintain the desired lower pressure. Inertia weights T and C prevent chattering.

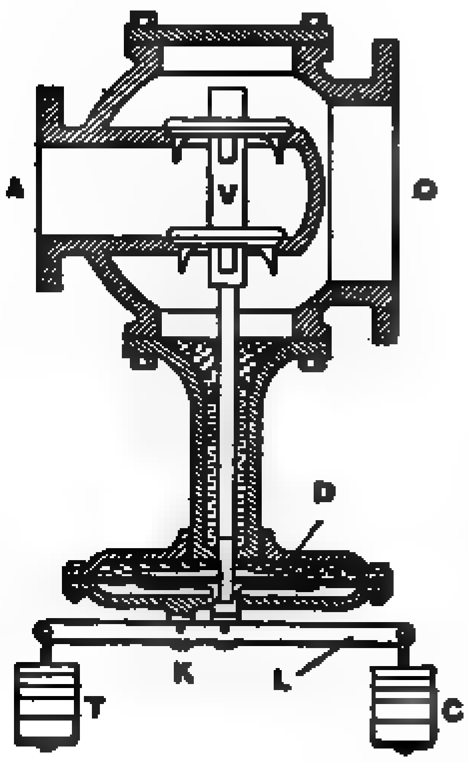


FIG. 396. Kieley Reducing Valve.

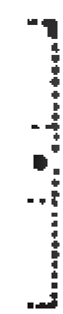
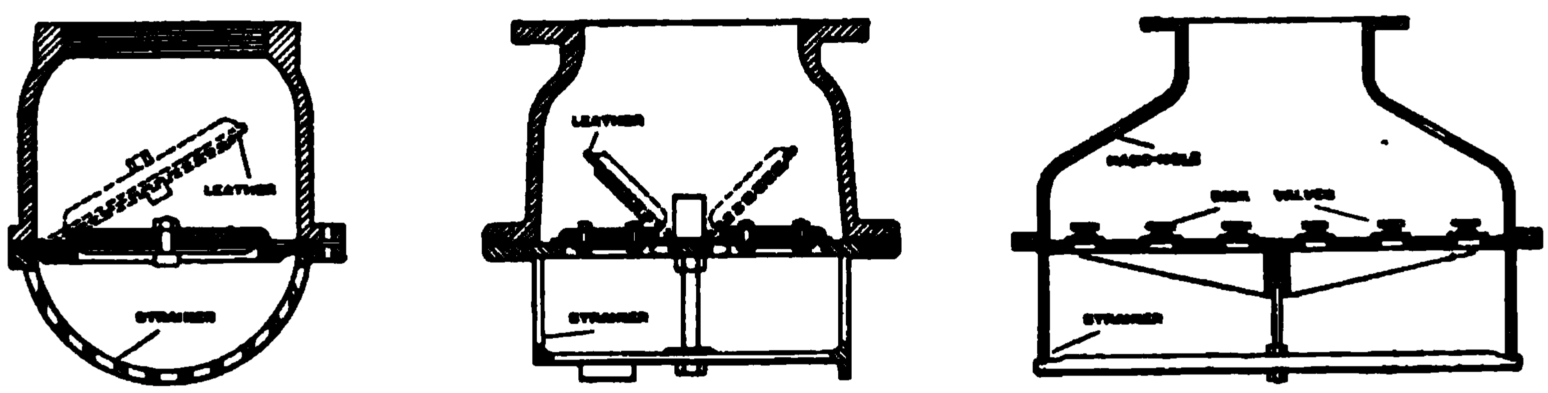


FIG. 397. Foster Pressure Regulator.

Fig. 397 shows a section through a class G Foster *pressure regulator* or reducing valve. In operation, steam enters at A and passes through the main valve port H to the outlet B. Steam at initial pressure passes through port C to chamber P and thence to the top of piston T through port L, opening the main valve U. Steam at delivery pressure passes through E and raises the diaphragm V against the pressure of spring R, allowing spring W to close the auxiliary valve X. The pressure in chamber J is then equalized by the reduced pressure in ports G and the under side of piston X, and thus allows spring Y to close the main valve, which is then held to its seat by the initial pressure. Any reduction in delivery pressure is transmitted to diaphragm V, and permits spring to open auxiliary valve X, thereby admitting steam to the top of piston T, as previously explained. The delivery pressure is adjusted by screw D; thus increasing the tension of spring R increases the discharge pressure and *vice versa*. The adjustment once made, the delivery pressure

will remain constant, regardless of any variable volume of discharge or of the initial pressure, so long as the latter is in excess of the delivery pressure. *W*, Fig. 366, shows the application of a reducing valve to an exhaust steam heating system. Live steam is led to the valve through pipe *A*. It will be noted that the pipe leading from the valve to the heating system is much larger than the high-pressure supply pipe on account of the increase in volume of the low-pressure steam. Reducing valves should always be by-passed to permit of repairs without shutting down the system. Care should be taken in not selecting too large a reducing valve, as the valve lift is very small and the larger the valve the less will be the lift for a given weight of flow and consequently the greater the wire drawing and erosion of the valve seat.

353. Foot Valves. — Whenever a long column of water is to be moved in either suction or delivery pipe it is customary to place a check valve near the lower end of the column to prevent the water from backing up when the pump reverses or shuts down. The check valve placed at the end of the suction pipe is called a *foot valve*. Any check valve may be used as a foot valve, though practice limits the choice to the disk or flap type as illustrated in Fig. 398. To prevent rubbish from destroying the action, a strainer or screen is gener-



(A) (B) (C)

FIG. 398. Types of Foot Valves.

ally incorporated with the body of the valve. *A*, Fig. 398, illustrates a *single-flap*, *B* a *multi-flap* and *C* a *disk* valve composed of a nest of small rubber valves. The single-flap are usually made in sizes ¾ to 6 inches, the multi-flap 7 to 16 inches, and the disk valve in all commercial sizes from ¾ to 36 inches. For large sizes, 16 to 36 inches, the multi-disk valve is given preference, since a number of the disks may be disabled without destroying its operation.

The Use and Abuse of Globe Valves: Power & Engr., Jan., 1909, p. 10.
Gate Valves in Steam Pipe Lines: Power & Engr., Feb. 16, 1909, p. 320.
Types of Check Valves and their Operation: Power & Engr., July 6, 1909, p. 11.

CHAPTER XVI.

LUBRICANTS AND LUBRICATION.

354. General. — The losses due to the friction of the working parts of machinery include considerably more than the mere loss of power, namely, the depreciation resulting from wear of bearings, guides, and other rubbing surfaces, and the expense arising from accidents traceable to excessive friction. The power absorbed in overcoming friction varies with the type of plant and the character of machinery and is seldom less than 5 per cent and often greater than 30 per cent of the total power developed. In large central stations these losses approximate 8 per cent and in weaving and spinning mills will average as high as 25 per cent. (Trans. A.S.M.E., 6–465.) These figures refer to properly lubricated plants operating under normal conditions. The proper selection of lubricant is therefore a very important problem, since, besides the cost of the lubricant itself, the loss in power and in wear and tear to machinery is no small item. A change of lubricant may frequently result in marked increase in economy of operation. The lubricants most commonly met with in power plant practice are conveniently classified as oils, greases, and solids, and are of animal, mineral, or vegetable origin.

Reference Books: Archbutt and Deeley, Lubrication and Lubricants; Redwood Lubricants; W. M. Davis, Friction and Lubrication; Gill, Oil Analysis; Robinson, Gas and Petroleum Engines; Thurston, Friction and Lost Work; Gill, Engine Room Chemistry.

355. Vegetable Oils. — Except for certain special purposes and for compounding with mineral oils these possess lubricating properties of little practical value, since they decompose at comparatively low temperatures and have a tendency to become thick and gummy. The vegetable oils sometimes employed are linseed, cottonseed, rape, and castor.

Vegetable Oils: Power, May, 1906, p. 300; Archbutt and Deeley, Lubrication and Lubricants, p. 232; W. M. Davis, Friction and Lubrication, p. 28; Gill, Oil Analysis.

356. Animal Fats. — Many animal fats have greater lubricating power than pure mineral oils of corresponding viscosity but are objectionable on account of their unstable chemical composition. They decompose easily, especially in the presence of heat, and set free acids

which attack metals. They are seldom used in the pure state and are usually compounded with mineral oils. The animal products used in this connection are tallow, neat's-foot oil, lard, sperm, wool grease, and fish oil, the first named being the most important. In cylinder lubrication, especially in the presence of moisture, the addition of 2 to 5 per cent of acidless tallow seems to make the oil adhere better to the metal surfaces and increases the lubricating effect, while the proportion is so small that ill effects from corrosion or gumming are scarcely perceptible.

Animal Fats: Archbutt and Deeley, Lubrication and Lubricants, p. 323; Gill, Oil Analysis, p. 44; Wright, Analysis of Oils, p. 193; Andes, Animal Fats.

357. Mineral Oils. — These are all products of crude petroleum and form by far the greater part of all lubricants. They present a wider range of lubricating properties than those derived from animal or vegetable sources, the thinnest being more fluid than sperm and the thickest more viscous than fats and tallows. They are not easily oxidized, do not decompose, become rancid, or contain acids.

Crude American petroleum of specific gravity 0.802 may yield the following commercial products. ("Gas and Petroleum Engines," W. Robinson.)

		Average Percentage.	Specific Gravity.	Boiling Point, Degrees F.
Light Oils.				
Petroleum ether	Cymogene	traces	0.590	32
	Rhigolene	0.1	.625–.631	64
	Gasoline	1–1.5	.635–.658	85–155
Petroleum spirit	C. Naphtha	10	.680–.700	140–212
	B. Naphtha	2–2.5	.717–.72	175–245
	A. Naphtha (benz.)	2–2.5	.742–.745	212–265
Burning oils, kerosene.	Water white	12–20	.780–.785	300–570
	Ordinary kerosene	40–55	.800–.810	300–680 and upwards
Fuel oils	For making oil gas; fuel	....	0.85	
Heavy oils	Lubricating oils	17.5	.885–.920	480 and upwards
	Paraffin wax	2	.908 at 60 deg. F.	
	Residium	5–10		

Mineral lubrication oils may be classified as

(1) *Distilled oils,* which are produced by distillation from crude petroleum and made pale, amber colored, and transparent by treatment with acid and alkali.

(2) *Natural oils*, which are prepared from crude petroleum, from which grit, suspended and tarry impurities have been removed. They are dark and opaque and are rich in lubricating properties.

(3) *Reduced oils*, or heavy natural oils, from which the lighter hydrocarbons have been evaporated and from which the tarry residue has been removed by filtration.

Mineral Lubricants: Engr. U.S., July 1, 1904, pp. 466, Vol. 44 (1907), pp. 241, 369, 542, 585; National Engr., Jan., 1905, p. 19; Eng. Mag., June, 1904, p. 455; Power, March, 1906, p. 146.

358. Solid Lubricants. — Dry graphite, soapstone, and mica are sometimes used as lubricants, though they are usually mixed with grease or oils. They cannot easily be squeezed or scraped from between the surfaces, and are consequently suitable where very great weights have to be carried on small areas and when the speed of rubbing is not high. The coefficient of friction of such lubricants is high, and when economy of power is essential better results may be secured by the use of liberally proportioned rubbing surfaces and liquid lubricants. Under certain conditions of pressure and speed these lubricants will sustain, without injury to the surfaces, pressures under which no liquid would work.

Graphite: Trans. A.S.M.E., 13–374; Engng., Aug. 16, 1907; Sci. Am., May 11, 1907; National Engr., Jan., 1904; Am. Mach., Dec., 1907, pp. 784, 934; Horseless Age, Jan., 1904, June 11, 1902, p. 712; Power, Dec., 1906, p. 758.

359. Greases. — Under this name may be included the various compounds which consist of oils and fats thickened with sufficient soap to form, at ordinary temperatures, a more or less solid grease. Those usually employed are lime, soda, or lead soaps, made with various fats and oils. "Engine" greases are thickened with a soap made from tallow or lard oil and caustic soda, and often contain neat's-foot oil, beeswax, and the like. For exceptionally heavy pressures, graphite, soapstone, and mica are sometimes added to the grease.

Greases: Jour. Eng. Soc. West. Penn., March, 1904, p. 112; Railroad Gazette, July 8, 1904, p. 131; St. Ry. Jour., July, 1905, p. 95; see also text-books given in references at beginning of chapter.

360. Qualifications of Good Lubricants. — A good lubricant should possess the following qualities:

(1) Sufficient "body" to prevent the surfaces from coming into contact under conditions of maximum pressure.

(2) Capacity for absorbing and carrying away heat.

(3) Low coefficient of friction.

(4) Maximum fluidity consistent with the " body " required.

(5) Freedom from any tendency to oxidize or gum.

(6) A high " flash point " or temperature of vaporization and a low congealing or " freezing point."

(7) Freedom from corrosive acids of either metallic or animal origin.

Lubricating oils are identified by certain tests which are used by refiners in grading and classifying the oils and by consumers in buying them. These tests usually cover the following:

(1) Identification of the oil, whether a simple mineral, animal or vegetable oil or a mixture.

(2) Density or gravity.

(3) Viscosity.

(4) Flash point.

(5) Burning point, fire test.

(6) Acidity.

(7) Coefficient of friction.

(8) Cold test.

361. Identification of Oil. — The chemical analysis of oils lies in the province of the chemist, but some of the characteristics may be readily determined by a few simple tests. To detect admixtures of fatty oils in mineral oil a small quantity is heated in a test tube for 15 minutes with small pieces of either metallic sodium or caustic potash. If fatty oil is present, saponification takes place and the soap formed will rise to the top as a semi-solid mass and the amount may be estimated. Tarry matter may be detected by dissolving a small quantity of oil in from 10 to 20 times its bulk of gasoline; the tar and other insoluble matter will separate and collect at the bottom.

Oil Testing and Specifications: Power, May, 1904, p. 302, Vol. 24 (1904), pp. 139, 240, 302, 526, Vol. 26 (1906), pp. 145, 222, 300, 331, 407; Am. Mach., April 11, 1907, p. 525; Engr. U.S., Oct. 15, 1904, p. 724, Oct. 2, 1905, p. 657; Marine Engng., June, 1903, p. 303; Chem. Engr. Nov., 1905, p. 10, Dec., 1905, p. 87, Jan., 1906, p. 141; Am. Gas Light Jour., Jan. 23, 1905; U.S. Cons. Repts., June, 1905; Sci. Am. Sup., Jan. 14, 1905.

362. Gravity. — The density or specific gravity is conveniently determined by means of a hydrometer, which, in the oil trade, is graduated according to the Baumé scale. The relationship between specific gravity and degrees Baumé at a temperature of 60 degrees F. may be expressed:

$$\text{Specific gravity} = \frac{140}{130 + \text{degrees Baumé}}.$$

Table 88 gives the specific gravity and gravity Baumé of a number of lubricating oils.

Gravity: Power, March, 1904, p. 139; Robinson, Gas and Petroleum Engines, p. 474; Archbutt and Deeley, Lubrication and Lubricants, pp. 172–185; W. M. Davis, Friction and Lubrication, p. 34.

TABLE 88.

SPECIFIC GRAVITY AND GRAVITY BAUMÉ OF A NUMBER OF LUBRICANTS.

	Specific Gravity.	Gravity Baumé.	Flash Test, Degrees F.
Water	1.000	10	....
Cylinder oil	.9090	24.5	575
Cylinder oil	.8974	26	540
Heavy engine oil	.9032	25.5	411
Medium engine oil	.9090	24	382
Light engine oil	.8917	27	342
Castor machine oil	.8919	27	324
Lard oil	.9175	23	505
Sperm oil	.8815	29	478
Tallow oil	.9080	24.5	540
Cottonseed oil	.9210	22	518
Linseed oil	.9299	19	505
Castor oil (pure)	.9639	15	....
Palm oil	.9046	25	405
Rape-seed oil	.9155	23	....
Spindle oil	.8588	33	312

363. Viscosity. — Viscosity may be defined as the degree of fluidity or internal friction of an oil. It is sometimes called the "body." It is determined by a viscosimeter. There are a number of different instruments for this purpose but no recognized standard instrument or method, so that "viscosity" conveys no meaning unless the name of the instrument, the temperature, and the amount of oil tested are given. Nearly all instruments are of the orifice type; that is, the viscosity of an oil is taken as the number of seconds required for a given amount, usually 50 cubic centimeters, to flow through an orifice at a given temperature. By "specific viscosity" is meant the ratio of the time required for the oil to run out to that of an equal quantity of water at 60 degrees F. The viscosity of engine oils is usually taken at 70 degrees F. and of cylinder oils at 212 degrees F.

Viscosity: Trans. A.S.M.E., 9–369; Engr., Lond., Sept. 7, 1906, p. 344, June 12, 1900, p. 633; Eng. Mag., June, 1907, p. 455; Machinery, May, 1903, p. 484; Power, May, 1904, p. 303, May, 1907, p. 293, March, 1906, p. 146.

364. Flash Point. — The flash point is determined by heating a sample of oil in an open or closed cup at the rate of 15 degrees F. per minute until a spark will ignite the vapor. The temperature at which

this occurs is the flash point. So much depends upon the extent of oil surface exposed, size of spark, distance spark is held from the oil at the time of ignition, and the dimensions of the cup, that there may be considerable variation in the flash point as obtained by different experimenters.

Flash Test: Power, April, 1906, p. 222; Robinson, Gas and Petroleum Engines, pp. 482–488; Archbutt and Deeley, Lubrication and Lubricants, pp. 187–191; W. M. Davis, Friction and Lubrication, p. 34; Gill, Oil Analysis, p. 36.

365. Burning Point, or Fire Test. — By continuing the application of heat and noting the temperature at which the oil takes fire and continues to burn, the burning point is obtained. The higher the temperature under which the oil must work the higher the fire test required, so that it will not decompose or volatilize. Too high a fire test gives an oil that does not atomize readily enough to reach all parts of the cylinder.

Consult references under " Flash Test."

366. Acidity. — The presence of free acid is determined by shaking up equal quantities of oil and water and testing with litmus paper. Another simple test is as follows: A small quantity of oil is placed in a test tube with a little cupric oxide (Cu_2O) and subjected to a gentle heat for three or four hours. The reaction with the copper turns the solution green if fatty acid is present and blue if vegetable acid is present.

Acidity: Power, April, 1906, p. 222; Archbutt and Deeley, Lubrication and Lubricants, pp. 215–218; Gill, Oil Analysis, p. 74.

367. Cold Test. — The " cold test " is the temperature at which the oil will just flow. The sample is solidified by means of a freezing mixture and the temperature noted when it softens sufficiently to flow.

Cold Test: Robinson, Gas and Petroleum Engines, p. 481; Archbutt and Deeley, Lubrication and Lubricants, pp. 195, 200–6; W. M. Davis, Friction and Lubrication, p. 28; Gill, Oil Analysis, p. 34; Redwood, Lubricants, p. 3; Power, March, 1906, p. 146.

368. Friction Test. — The coefficient of friction as determined from friction-testing machines is useful in obtaining a comparison of oils under the test conditions, but gives little information concerning the action of the oil under the widely different conditions found in actual practice. Table 89 gives the physical properties of a number of lubricating oils, with their particular zone of application.

Friction and Lubrication: Trans. A.S.M.E., 1–74, 6–136; Am. Mach., July 21, 1904, p. 956, Jan. 23, 1902, p. 113, Sept. 10, 1903, p. 1303; Am. Elecn., Nov., 1905, p. 557; Engr., Lond., June 19, 1903, p. 631; Power, Dec., 1905, p. 748; National Engineer, Jan., 1905, p. 19; Mech. Engr., Nov. 30, 1907; Pro. Inst. Civ. Engr., 1901, p. 146; Machinery, Aug., 1903, p. 631; Proc. A.S.M.E., Nov., 1909, p. 1099.

369. Atmospheric Surface Lubrication. — In a general sense all journals, slides, and "atmospheric" surfaces should be lubricated with straight mineral oils (as free from paraffin as possible), except when in contact with considerable water, in which case it is advisable to add 20 to 30 per cent of lard oil. Vegetable oils, paraffin oils, and animal oils (except lard oil as above stated) are not recommended for general engine and dynamo service. The test requirements of a number of classes of lubricants are outlined in Table 89 and represent current practice. Bearings, guides, and all external rubbing surfaces may be lubricated in a number of ways. (1) They may be given an *intermittent* application of oil, as, for example, with an oil can; (2) they may be equipped with oil cups with *restricted* rates of feed; and (3) they may be *flooded* with oil. The relative lubricating values of the systems have been estimated approximately as follows (*Power*, December, 1905, p. 750):

	Coefficient of Friction.	Comparative Value.
Intermittent	0.01 and greater	72 and less
Restricted feed	0.01 to 0.012	79 to 86
Flooded bearing	0.00109	100

370. Intermittent Feed. — Intermittent applications are ordinarily limited to small journals, pins, and guides which are subject to light pressures and which do not easily permit of oil or grease cups, as, for example, parts of the valve gear of a Corliss engine, governors and link work. On account of the labor attached and the frequent doubt about the oil reaching the wearing surfaces this method of lubrication is limited as much as possible even in the smallest plants.

371. Restricted Feed. — In the average power plant the major part of the lubrication is effected by means of oil cups which are filled at intervals by hand or by mechanical means, the oil being fed from the cup by drops, according to the requirements.

372. Oil Bath. — In large power plants the principal journals and wearing parts are supplied with a continuous flow of oil which completely "floods" the rubbing surfaces. The oil is forced to the various parts either by gravity from an elevated tank or by pressure from a pump. After the oil leaves the bearings it flows into collecting pans, thence into a receiving and filtering tank, and finally is pumped back into an elevated reservoir and used over and over again. The little lost by leakage and depreciation is replenished by the addition of new oil to the system.

TABLE 89.

PHYSICAL CHARACTERISTICS OF A NUMBER OF LUBRICANTS.

(*Power*, December, 1905, p. 750.)

Kind of Oil.	Use and Adaptation.	Gravity, Degrees.	Cold Test, Degrees.	Flash Test, Degrees.	Fire Test, Degrees.	Viscosity at 70 Degrees.
High-pressure cylinder oil.	For steam cylinders using dry steam at pressures from 110 to 210 pounds.	25 to 24.5	30	600 to 610	645 to 660	175 to 205
General cylinder oil . .	For steam cylinders using dry steam at 75 to 100 pounds. For air compressor cylinders when made from steam-refined mineral stock and when viscosity is 200.	26 to 25.5	30	550 to 585	600 to 630	180 to 190
Wet cylinder oil. (Remark 1.)	For use where the steam is moist, especially in compound and triple expansion engines.	25.8 to 25.3	30	560 to 585	600 to 630	150 to 185
Gas engine cylinder oil. (Remark 2.)	For gas engine cylinders. Neutral mineral oil compounded with an insoluble soap to give body.	26.5	30	320	350	300
Automobile gas engine oil. (Remark 3.)	For automobile gas engines and similar work.	29.5	30	430	485	195
Heavy engine and machinery oils.	For heavy slides and bearings, shafting, and horizontal surfaces.	30.5 to 29.5	30	400	440 to 450	170 to 195
General engine and machine oils.	For high-speed dynamos and machines.	30.8 to 30	30	400 to 420	450 to 470	175 to 190
Fine and light machine oils.	For fine work, from printing presses to sewing machines and typewriter oils. With a cold test of 25° to 28° and a viscosity of 140° this makes an excellent spindle oil.	32.5 to 30.2	30	400	440	110 to 160
Cutting and heat dissipating oils. (Remark 4.)	For cutting tools, screw cutting and similar work.	27 to 23	30	410 to 420	475 to 480	210 to 175
Refrigerating oils.	For ice machinery.	30.2	0	200	225	165
Wet service and marine oils. (Remark 4.)	For marine service, or where a great deal of moisture must be handled.	28	30	430	475	230
Greases.	They are used in special work and for heavy pressures moving at slow velocities.					

Remark 1. — May contain not over 2 to 6 per cent of refined acidless tallow oil in the high-pressure oils and not over 6 to 12 per cent in the low-pressure oils.

Remark 2. — The reason for using an insoluble soap such as oleate of aluminum is that it is impossible to decompose the soap with a high heat; the soap, although not a lubricant, is a vehicle for carrying some oil.

Remark 3. — Owing to a lack of body, this oil will not interfere with the sparking by depositing carbon on the platinum point.

Remark 4. — May contain 30 to 40 per cent of pure strained lard oil.

373. Oil Cups. — Fig. 399 illustrates the application of *sight-feed* oil cups to the crosshead and slides of a reciprocating engine. The oil is fed into the cups by hand and gravitates to the rubbing surfaces, the rate of flow being regulated by a needle valve. Cups *A* and *B* feed directly to the crosshead guides, but the oil from cup *D* flows to the bottom orifice *O*, from which it is wiped by a metallic wick *S* and carried by gravity to the wrist pin.

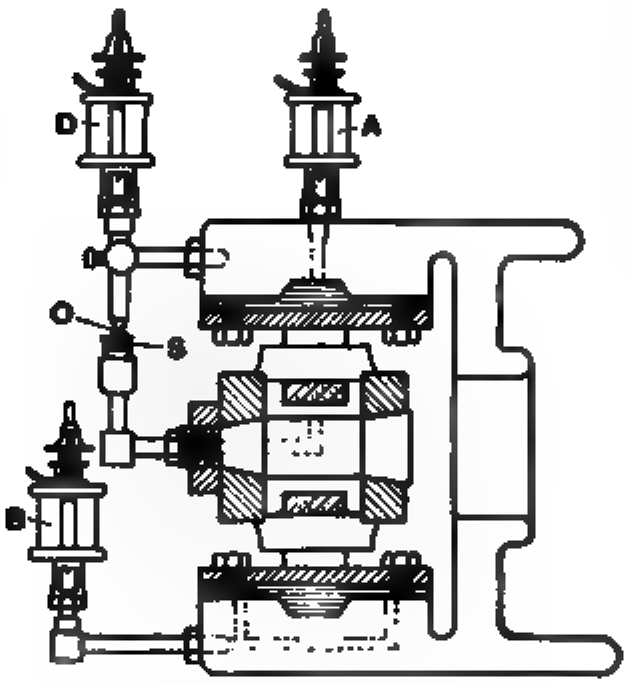


FIG. 399. Oil Cup Lubrication, Hand Filled.

374. Telescopic Oiler.—Fig. 400 shows the application of a *telescopic oiler* to a crosshead and guides. *O* and *C* are sight-feed oil cups, the former feeding directly to the top guide through the tube *S*. The oil from *C* flows by gravity through the swing joint into the telescopic tubes *P*, *R* and thence to the pin through the lower swing joint as indicated. As

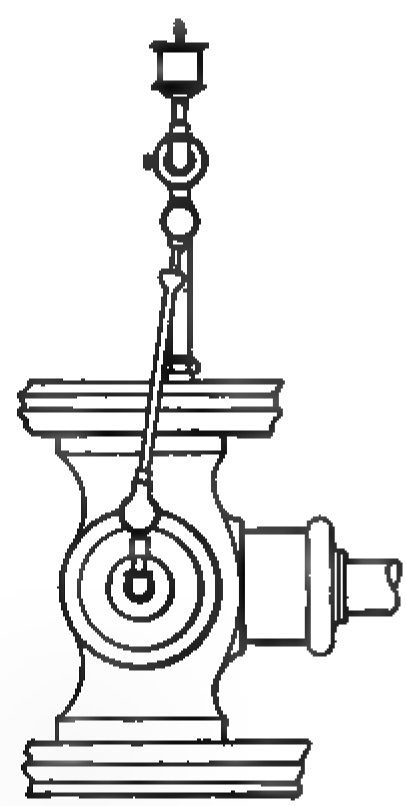


FIG. 400. Nugent's Telescopic Oiler.

the crosshead moves back and forth, the pipe *P* slides into and out of pipe *R*, the oil being thus conducted directly to the pin without wasting.

A device of this type installed on a high-speed automatic engine at the Armour Institute of Technology has been in operation for three years without cost for repair or renewal.

375. Ring Oiler. — Small high-speed engines are often oiled by the *oil-ring* system, as illustrated in Fig. 401. The shaft is encircled by

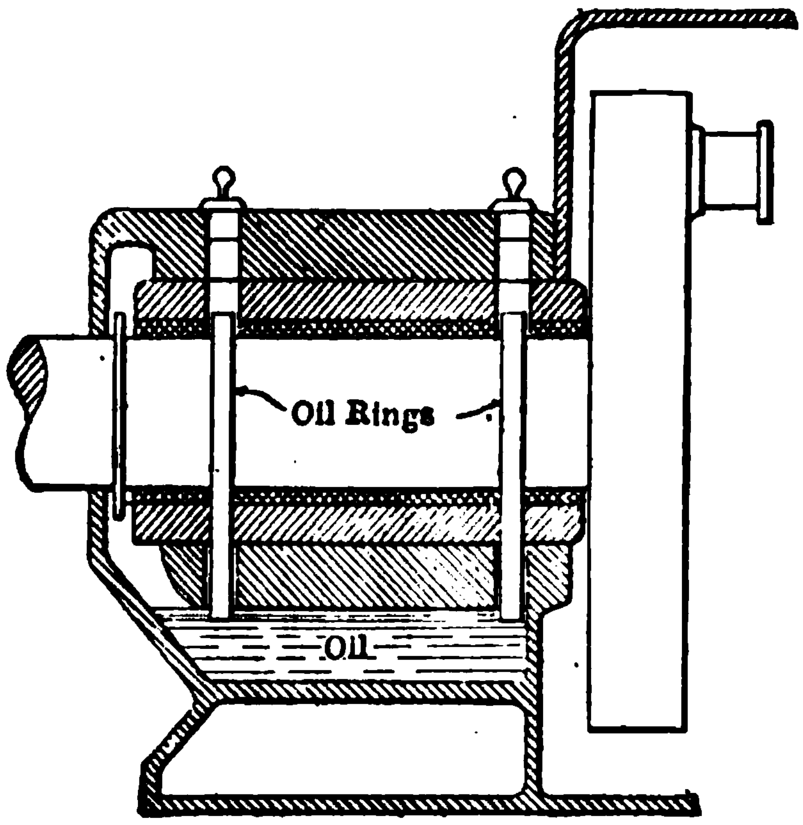


Fig. 401. Oil Ring Lubrication.

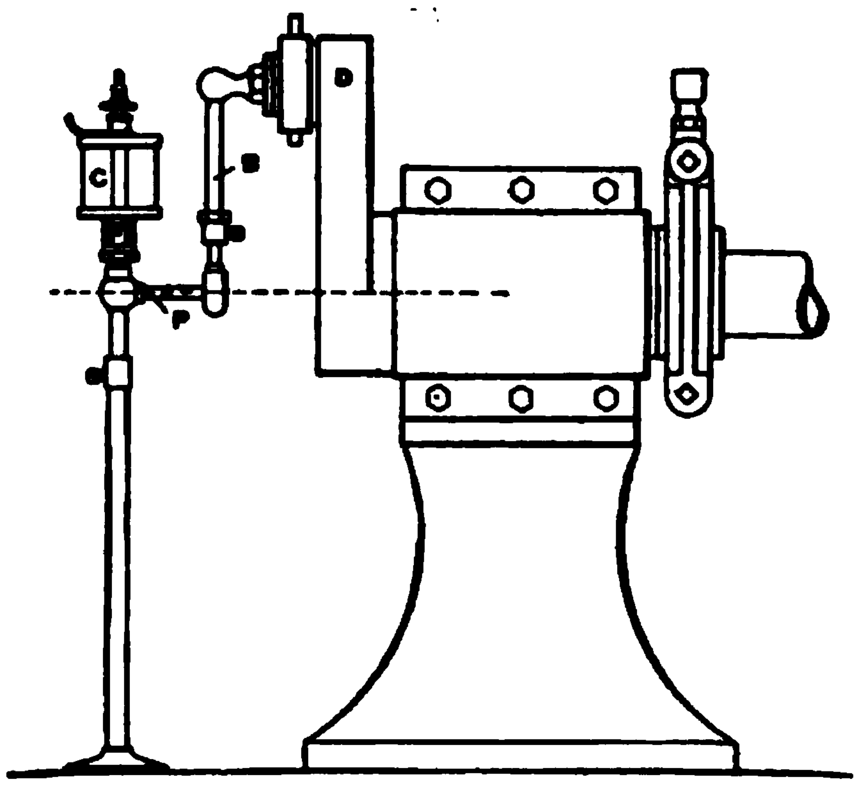


Fig. 402. Centrifugal Oiler.

several loose rings which dip into a bath of oil in the base of the pedestal or frame and, rolling on the shaft as it turns, carry oil to the top of the shaft where it spreads to the bearings. In some cases the rings are replaced by loops of chain.

Ring Lubrication: Zeit. d. Ver. Deutscher Ing., Aug. 10, 1907.

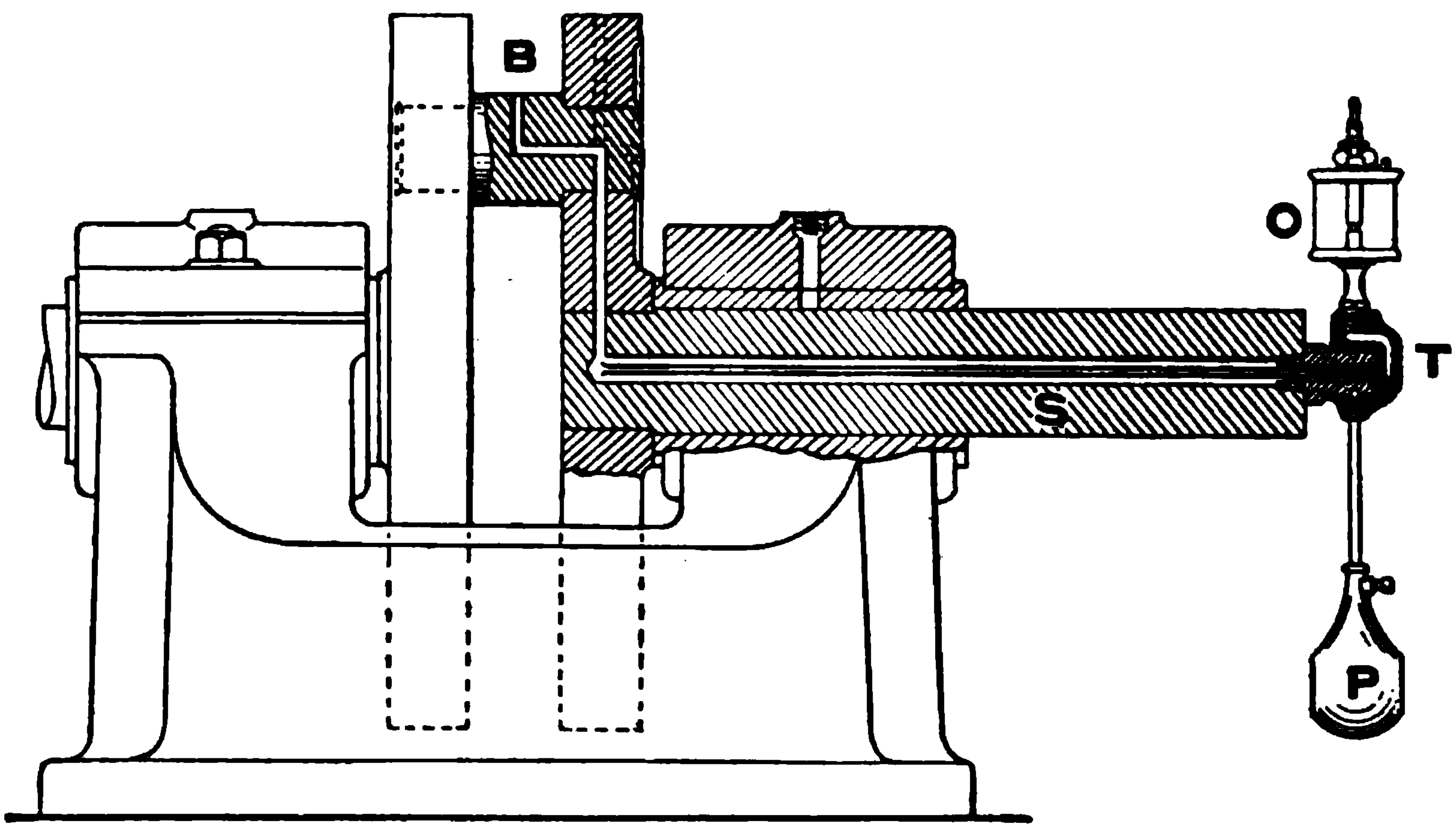


Fig. 403. Pendulum Oiler.

376. Centrifugal Oiler. — Fig. 402 illustrates the application of a *centrifugal oiler* to a side-crank engine. The oil supply is regulated by

the sight-feed cup C and flows by gravity to the pipe P in line with the center of the crank shaft. Centrifugal force throws the oil outward through pipe B to the center of the pin D, which is drilled longitudinally and radially so as to distribute the oil upon the bearing surface.

377. Pendulum Oiler. — Fig. 403 illustrates the application of a *pendulum oiler* to the crank pin of a center-crank engine. Oil cups and pendulum P are fastened to the crank shaft S by trunnion T. The pendulum holds the cup vertical, since the friction of the trunnion is not sufficient to revolve it. Oil flows along the center of the crank shaft under the head of oil in cups O and is thrown outward to bearing B by centrifugal force.

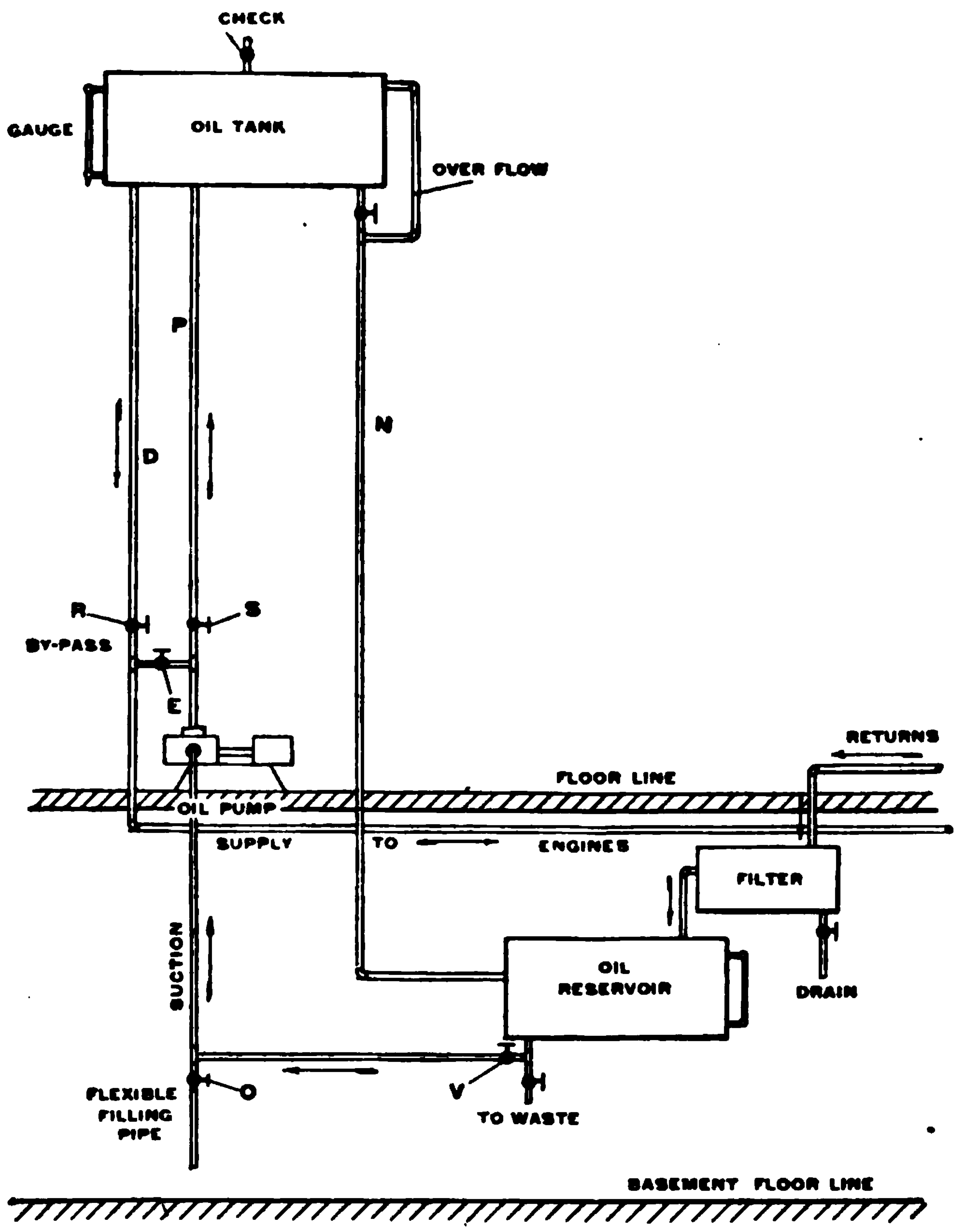


FIG. 404. Simple Gravity Feed System.

378. "Splash" Oiling. — In some high-speed engines the crank, connecting rod, and crossheads are inclosed by a casing, the bottom of

which is filled with oil to such a depth that at each revolution of the crank, the end of the connecting rod is partly submerged. The result is that the oil is splashed into every part of the chamber, and the crank pin, crosshead pin, and crosshead slides practically run in an oil bath.

379. Gravity Oil Feed. — Fig. 404 illustrates a simple *gravity oil feed* system. The oil to the engine is supplied from the oil tank by pipe *D* under pressure corresponding to the height of the tank above the oil cups. After performing its function the oil gravitates to the filter and from the latter to the oil reservoir, from which it is pumped back to the supply tank, the overflow being returned to the reservoir through pipe *N*. Operation is interrupted only when new oil is to be added to the system from the barrel through the flexible filling pipe. In case the oil tank is put out of commission, or the supply pipe becomes clogged, full pump pressure may be used by closing valves *R* and *S* and opening valve *E*. The make-up oil is small in amount compared to the quantity circulated. The reclaiming and purifying of the oil are essential if the bearings are to be flooded, otherwise the cost of oil would be prohibitive. At the power house of the South Side Elevated Railway the daily circulation (24 hours) of engine oil is approximately 1500 gallons. The make-up oil amounts to eight gallons.

An objection sometimes made to the above system is that the varying heights of oil in the supply tank may cause considerable variation in pressure at the oil cups, causing them to feed faster when the tank is full and slower when the tank is nearly empty. This applies only to installations where the supply tank is filled intermittently.

380. Low-Pressure Gravity Feed. — Fig. 405 shows the application of a low-pressure oiling system in which the level in the sight feeds is kept constant. *A* is the main supply tank, B^1 and B^2 the upper and lower gauges indicating the oil level, *C* the supply pipe running to the engines, and *D* a small standpipe closed at one end and vented near the top. The reservoir is supplied with oil by the valve marked " inlet." When the tank is filled the oil rises in the standpipe *D* a corresponding height. The inlet valve is then closed and the oil in the standpipe feeds down to the level of the sight feeds or to a point where the air will enter the bottom of the tank. This will be the constant oil level, since oil flows from the tank only in proportion to the amount of air admitted. A head of 6 inches has been found to give the best results. (*Engineer*, U. S., March 16, 1903, p. 243.)

381. Compressed-Air Feed. — Fig. 406 shows diagrammatically the arrangement of the oiling system at the First National Bank Building, Chicago. The storage tank containing the supply of engine oil is

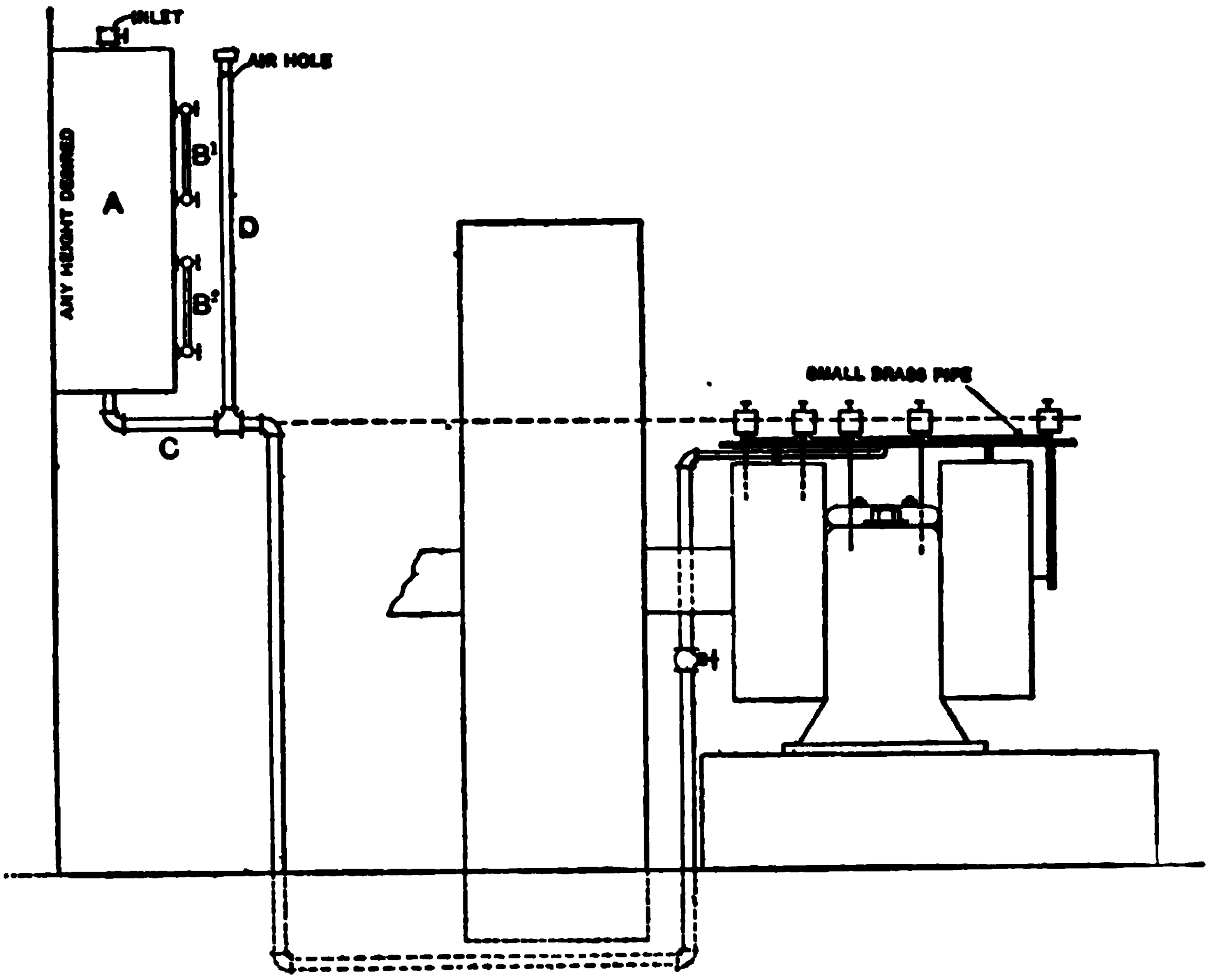


FIG. 405. Low-Pressure Gravity Feed, Constant Head.

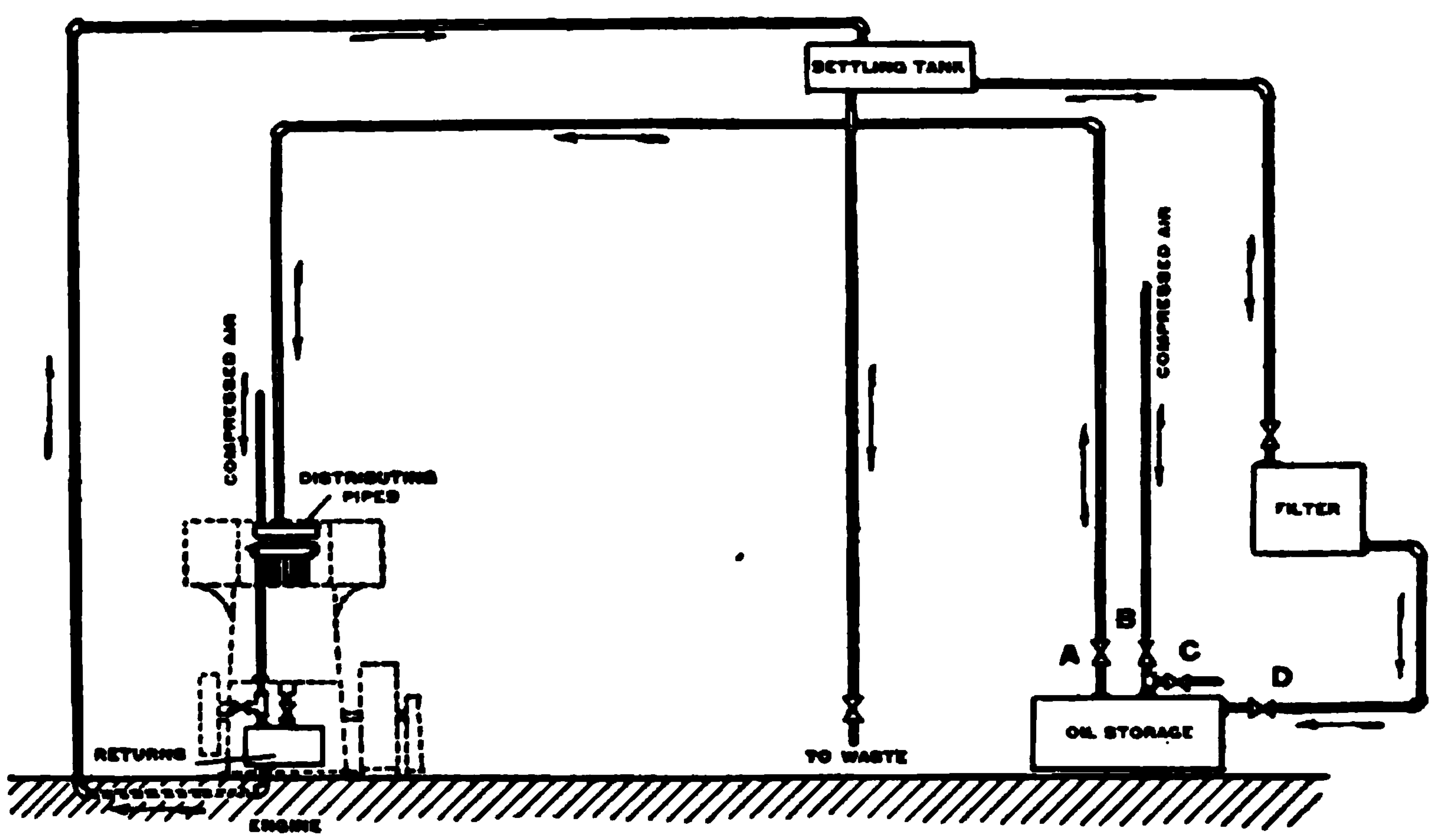


FIG. 406. Oiling System at the Power Plant of the First National Bank Building, Chicago

under air pressure at all times except during the short periods when it is being filled with oil from the filter. The air pressure on the surface of the oil forces it to a manifold on the engine from which it is distributed to the various oil cups. The oil flows from the different bearings to the returns tank located at the base of the engines. When the tank is filled air pressure is admitted and the oil forced to the settling tank, which has a capacity of about 400 gallons and is located near the ceiling. The oil is allowed to settle and the entrained water and foreign material are drained to waste. The oil gravitates from this tank to a series of Turner oil filters. When a new supply of oil is needed, valves *A* and *B* are closed and vent valve *C* opened, cutting off the supply of air and reducing the pressure to atmospheric. Valve *D* is then opened and oil flows from the filters to the storage tank.

Lubricating Systems. — *Lubrication of Line and Counter Shafting:* Trans. A.S.M.E., 10–810. *Gravity Oil Systems:* Power, Nov., 1902, p. 23, July, 1906, p. 409, June, 1903, p. 305. *Oiling Systems for Electric Engines:* Elec. World, July 7, 1906, p. 26. *Oiling System for Power Plants:* Engr. U.S., March 16, 1903, p. 243, April 15, 1904, p. 278; National Engr., Feb., 1905, p. 10, March, 1905, p. 16.

382. Cylinder Lubrication. — The test requirements for cylinder oils are outlined in Table 89, from which it will be seen that pure mineral oil fulfills practically all requirements for dry steam. In connection with moist steam, as in the low-pressure cylinders of compound engines, an addition of from 2 to 5 per cent of acidless tallow oil is recommended. Vegetable oils, beeswax, lard oil, degras (wool grease), and the like should never be used in compounding cylinder oils. The best cylinder oils are made from Pennsylvania stock.

Cylinder oils must be forced to the parts requiring lubrication against the prevailing steam pressure, which is ordinarily accomplished by (1) *cylinder cups,* (2) *hydrostatic lubricators,* or (3) hand or power driven *force pumps.*

383. Cylinder Cups. — A cylinder oil cup consists essentially of a steam-tight brass vessel fitted at the bottom with a pipe connection and valve. A screwed cap offers a means of introducing the lubricant into the cup. After the cap is in place the valve is opened and the cup is subjected to full steam pressure. The pressure in the cup being equal to that in the steam chest or cylinder, permits the lubricant to gravitate through the valve into the cylinder.

Fig. 407 shows a section through an improved form of oil cup in which the oil feeds from the top instead of the bottom as is the case with the common form of cylinder cup. The vessel is attached to the steam chest or to the supply pipe below the throttle valve. Steam is admitted through opening *B* and, condensing, settles through the oil

to the bottom. This raises the level of the oil until it begins to overflow down the same passage by which the steam enters. This action is intensified by the fluctuation in steam pressure. The rate of feeding is regulated by valve *C* and tested by unscrewing plug *F*. If oil appears through opening *G*, the cup is feeding oil; if steam or water is emitted, the cup is empty. The cup is filled by means of plug *E* and the water drained at *D*.

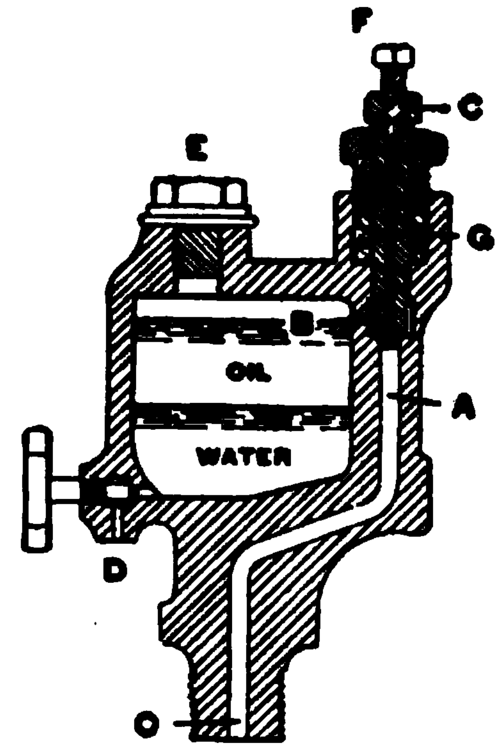


FIG. 407. Leyland Automatic Cylinder Cup.

384. Hydrostatic Lubricators. — The most common method of cylinder lubrication is by means of *hydrostatic* lubricators of the sight-feed class, Fig. 408. The principle of operation is as follows: The lubricator is filled with cylinder oil by removing cap *K*, the height of oil appearing in glass *L*. If water is present the oil floats on top as indicated. After the cap is screwed in place the valves in the condenser pipe are opened, subjecting the oil in the vessel to steam-pipe pressure. Steam is condensed in pipe *C*, filling tube *B* and part of *C*, thus adding to the steam pressure the pressure due to the weight of the water column. Valve *F*, which communicates with the top of the vessel by means of tube *A*, is opened wide, as is also the regulating valve *I*. The pressure at *B* being greater than that at *A* by an amount equivalent to the height of the water column, forces the oil through *A* and the "sight feed" *S* to the steam pipe. The rate of flow is controlled by the regulating valve *I*. As the oil flows from the vessel its space is occupied by condensed steam, the height of oil and water being visible in glass *L*. Owing to the small capacity of the lubricator it must be refilled frequently. To reduce ~~the~~ amount of labor required with the above apparatus, independent sight feeds, Fig. 409, are sometimes used in connection with a central reservoir. Such an installation is shown diagrammatically in

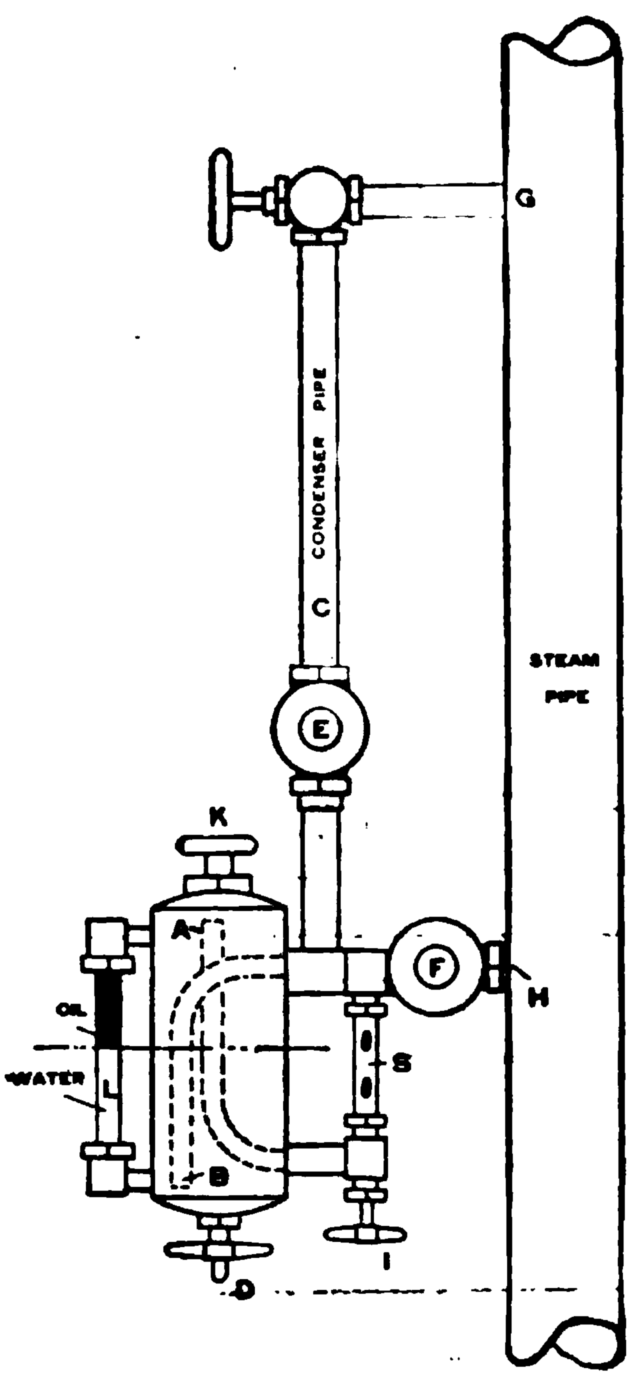


FIG. 408. Common Hydrostatic Lubricator.

Fig. 410. A condenser pipe leading from the steam main enters the bottom of the reservoir and the condensed steam fills up the reservoir as fast as the oil is fed out. The principle is the same as that of the simple hydrostatic lubricator. Oil is frequently injected by mechanical means under a steady pressure generated and governed independently of the steam. Two systems are in common use, direct mechanical pump pressure and air pressure.

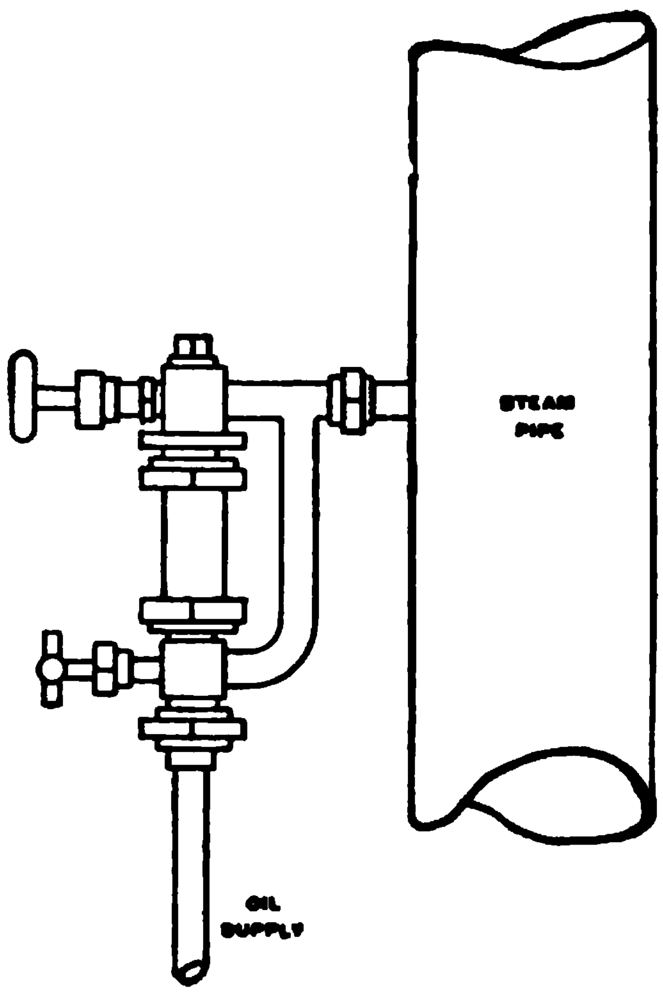


Fig. 409. Lunkenheimer Sight-Feed Lubricator.

385. Forced-Feed Cylinder Lubrication. — Fig. 411 illustrates the "Rochester" simple feed automatic lubricating pump, which takes the oil by gravity from the reservoir through a sight-feed glass and forces it through a small pipe to the steam supply pipe. The pump entirely obviates the trouble due to intermittent feeding and, being directly driven from the engine, runs at constant speed. The feed is uniform and independent of the pressure pumped against. The rate is determined by the length of stroke of the pump piston, which is easily adjusted.

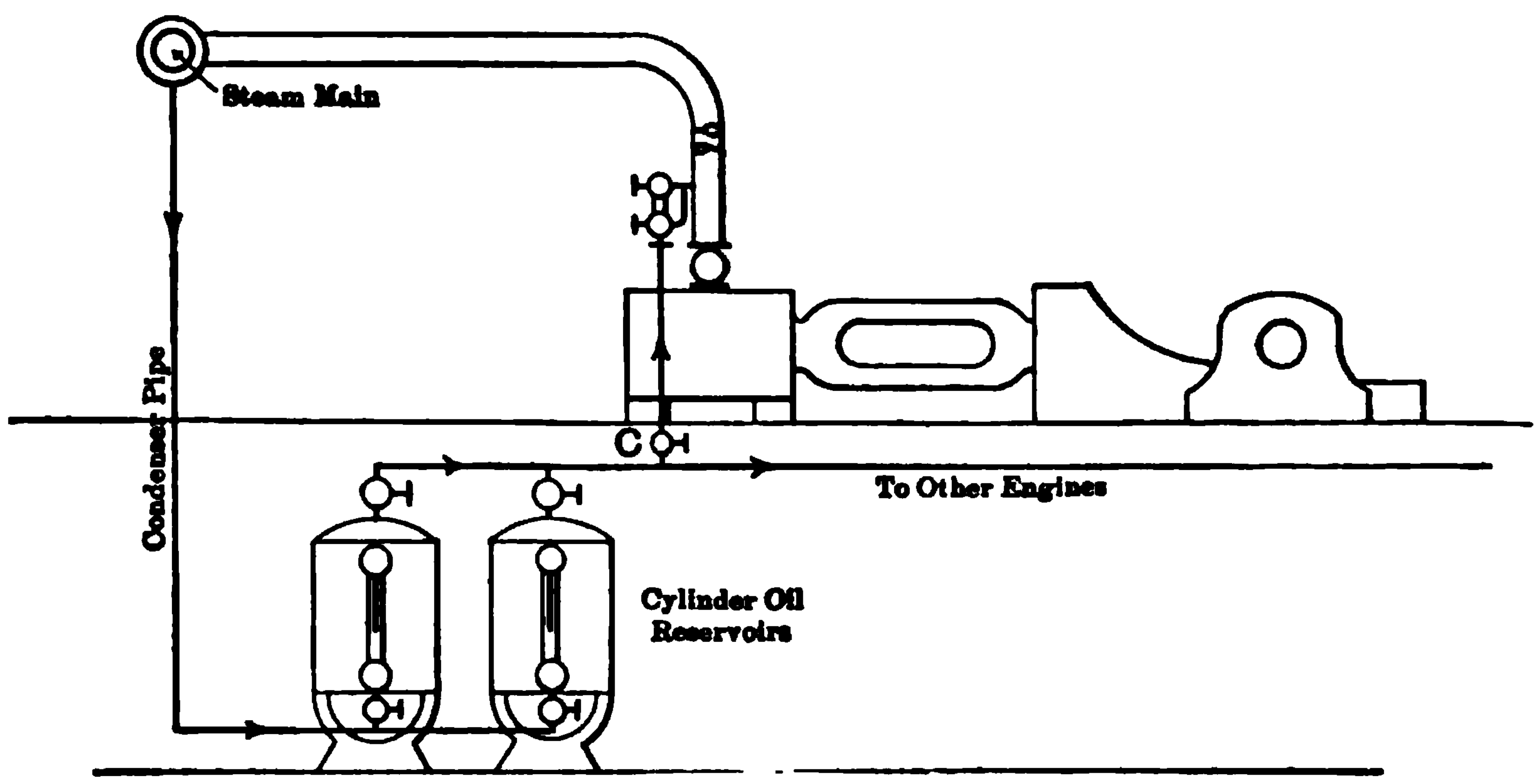


Fig. 410. Central Hydrostatic Lubricator.

With large engines multi-feed pumps are sometimes used, which force oil to the various valves as well as to the steam pipe. Fig. 412

shows an arrangement of storage tank in connection with pump reservoir to avoid the trouble of hand filling.

FIG. 411. Rochester Forced-Feed Lubricator.

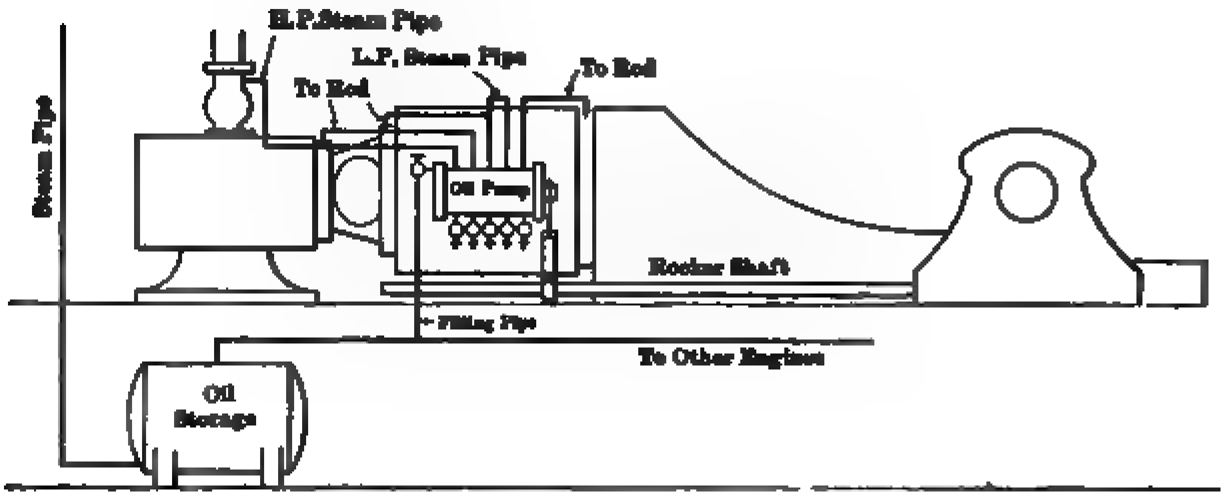


FIG. 412. Forced-Feed Cylinder Lubrication.

386. Siegrist System. — Fig. 413 shows an application of the Siegrist system of cylinder and engine lubrication. There are two storage tanks on the engine-room floor, one for cylinder oil and the other for engine oil, the distributing arrangements being the same in each case. The oil is pumped from each tank into a main pipe extending the length of the engine room and provided with branches at each point requiring lubrication. The oil pumps are actuated by steam and are of the duplex direct-acting type, provided with automatic governors which regulate the speed to suit the demand for oil.

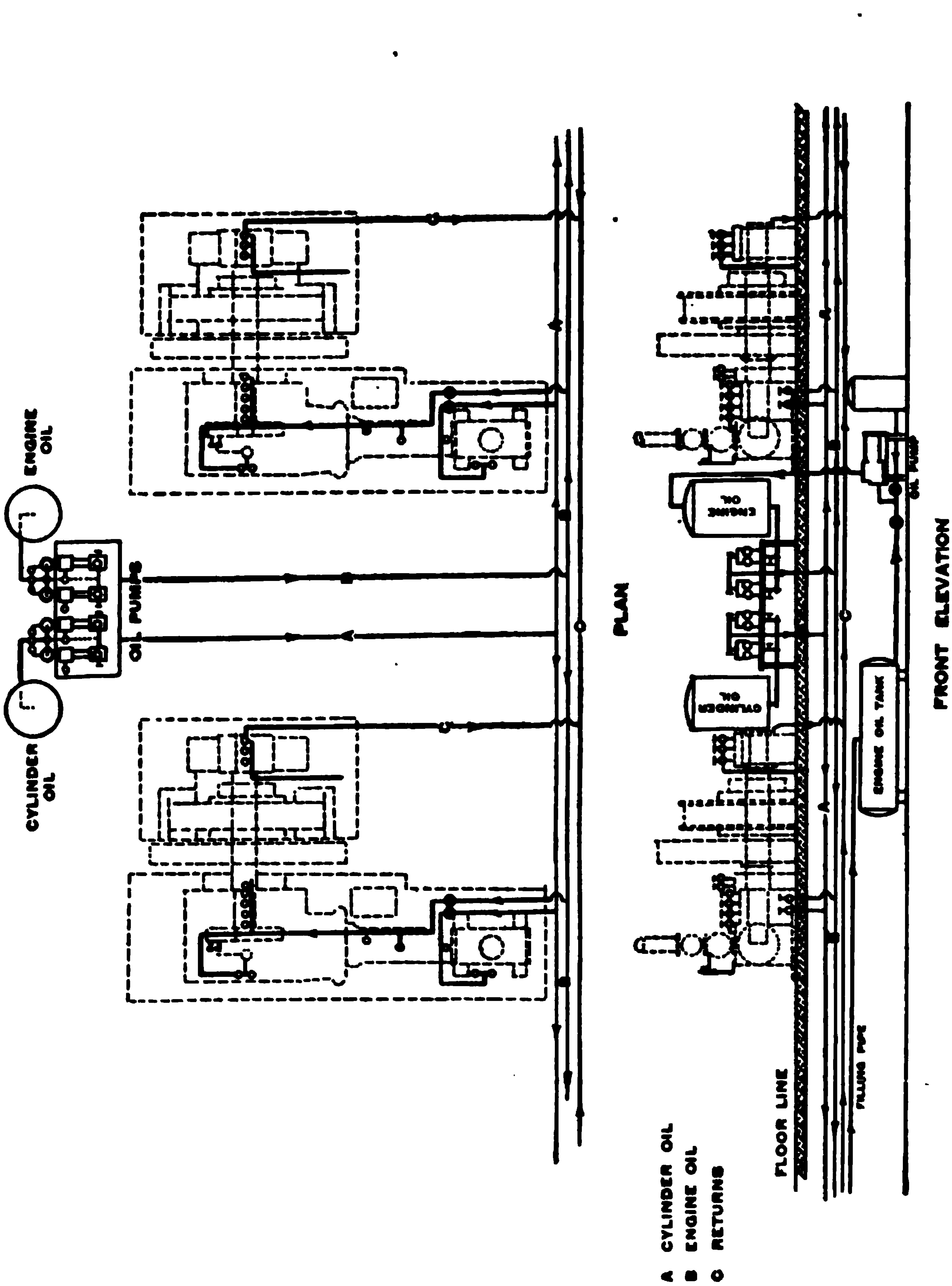


FIG. 413. Siegrist System of Lubrication.

The cylinder oil is forced through a special sight-feed lubricator, Fig. 407, under a pressure of about 25 pounds in excess of the steam pressure. Referring to Fig. 414, diaphragm valve D, in the bottom of the lubricator, is kept closed by the steam pressure admitted through pipes B. Thus the inlet pressure must be greater than that of the steam before the valve will open and admit oil to the engine. The oil, after entering, passes upward through the sight-feed glass and downward through the hollow arm A to the steam pipe. The engine oil is forced by the pump to the various points under a pressure of about 20 pounds. The waste oil is caught in suitable receptacles and, after being filtered, is returned to the storage tank by a steam pump. This pump is connected so that it can supply the storage tank either from the filter or with fresh oil from a large oil tank in the basement. By this arrangement all handling of oil in the engine room is done away with.

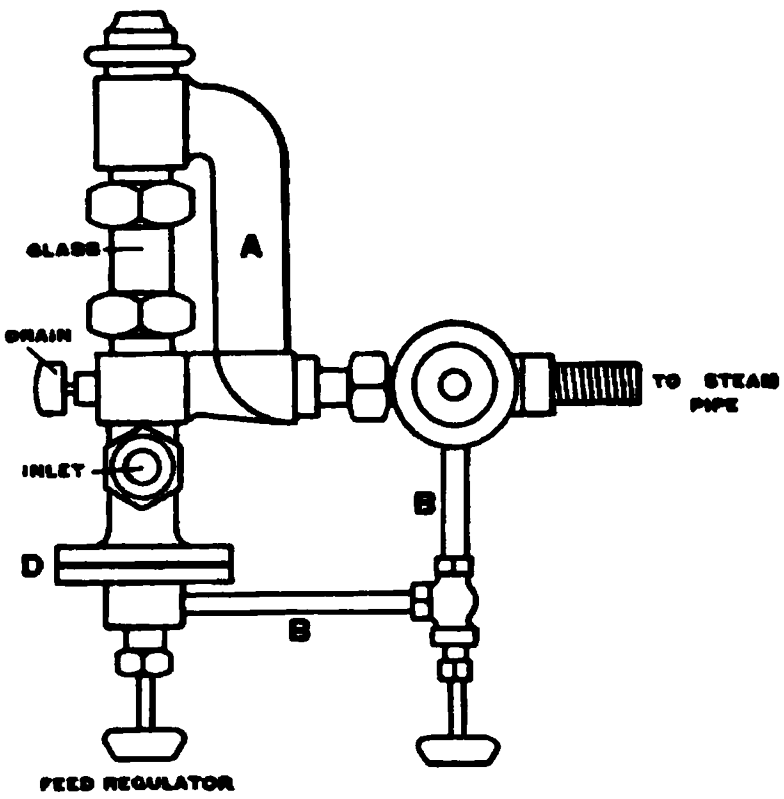


FIG. 414. Siegrist Sight-Feed Lubricator.

387. Oil Filters. — After oil has been applied to machinery its lubricating properties become impaired on account of (1) contamination with anti-lubricating material, such as dust, metallic particles from wear, gum, acid, and resin; and (2) exposure to heat and the atmosphere which drives off part of the more volatile constituents and decreases the fluidity of the oil.

In many small plants no attempt is made to reclaim oil that has once been used, since the quantity is so small that the cost and trouble involved would more than offset the gain. Where large quantities of oil are used, considerable saving may be effected by using it over and over again. To render the oil fit for reuse it must be thoroughly purified. The anti-lubricating matter is removed by precipitation and filtration.

Fig. 415 shows a section through a "White Star" oil filter and purifier. The apparatus consists of a cylindrical sheet-iron vessel divided into two compartments by a vertical partition. These two compartments are connected near the top by valve B. The smaller chamber is provided with a funnel A and a steam coil for heating the contents. The large chamber contains a cylindrical wire screen covered with several folds of filtering cloth. Impure oil is poured into funnel A, the upper part of

which is provided with a removable sieve or strainer, and is discharged below the surface of the water through holes in the foot of the tube. The thin streams of oil rise vertically to the surface of the water and the heavy particles of grit and dirt gravitate to the bottom. The steam coil heats the oil and water and facilitates precipitation of the solid matter by thinning out the streams of oil. When the oil in the smaller chamber reaches the level of valve *B* it flows into the filter bag, which removes the remaining impurities and permits the purified products to flow into the large compartment from which it may be drawn at will. All parts are accessible and readily removed for cleaning purposes. The

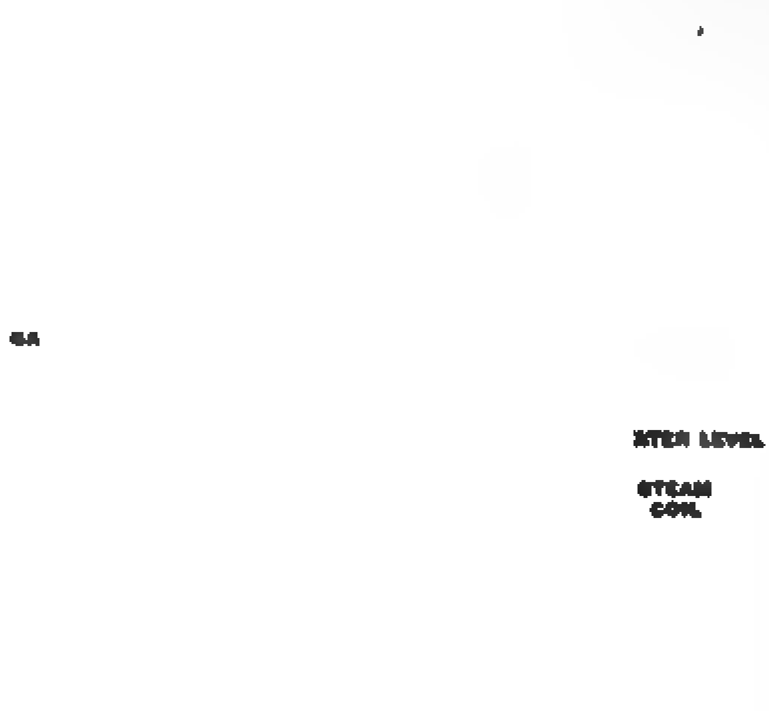


Fig. 415. White Star Oil Filter.

accumulated sediment in the bottom of the small chamber is discharged to waste at intervals by means of a suitable drain. When the

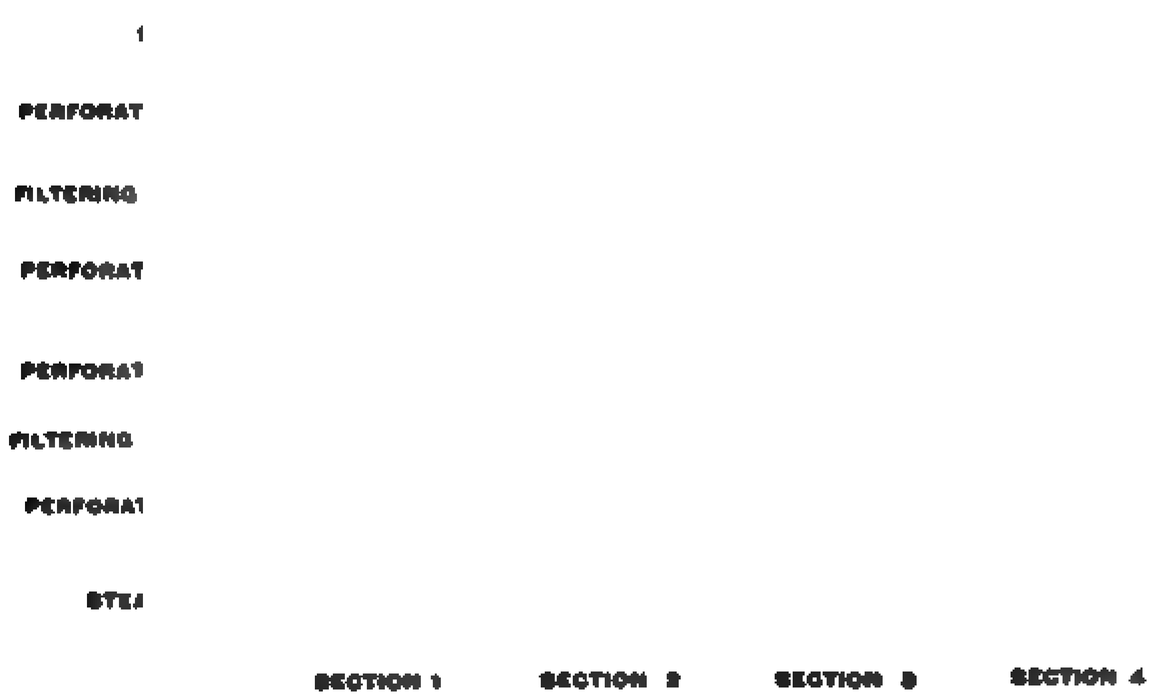


Fig. 416. Turner Oil Filter.

filter cloth is to be removed, valve *B* is closed and the wire cylinder is disconnected and lifted out. Any oil remaining in the filter is returned to funnel *A*. The filter cloth is held against the screen by cords and hence is readily removed.

Fig. 416 shows a section through a Turner oil filter, illustrating the type of filter usually installed in large stations where continuous filtration is desired. This apparatus consists of a rectangular tank divided into four compartments. The returns from the lubricating system flow into section 1 through a screened funnel and discharge into the water space at the bottom of the compartment. The oil rises through the water, passes, under pressure of the head in the funnel, through a layer of filtering material resting on a perforated plate, and collects in an inverted cone. Through perforations round the top of the cone it passes into a dirt chamber, where most of the heavy impurities are deposited, and then, still rising, passes through another perforated plate and more filtering material. The partially cleaned oil, which issues, overflows into the second compartment and thence into the third, the same cycle of operations being repeated in these two. The overflow from the third compartment descends through a final filter in the fourth compartment and collects at the bottom, from which it is withdrawn by the oil pump.

Forced-Feed Lubrication: Am. Elecn., Aug., 1902, p. 402, Dec., 1905, p. 608; Automobile, Nov. 1, 1906, p. 572; Mech. Engr., April 20, 1907, p. 552.

Cylinder Lubrication: Power, Dec., 1902, p. 30, Jan., 1905, p. 36, March, 1906, p. 163; Engr., Lond., 1905, Vol. 96, pp. 55, 108, 132, 155; St. Ry. Jour., June 22, 1907, p. 1103; Engr. U.S., Oct. 15, 1906, p. 682; Am. Gas Light Jour., Jan. 23, 1905, p. 130; Horseless Age, Sept. 24, 1902, p. 676.

Miscellaneous. — Measurement of Durability of Lubricants: Trans. A.S.M.E., 11-1013. *Valuation of Lubricant by Consumer:* Trans. A.S.M.E., 6-437. *Suitability of Lubricants:* Power, Nov., 1906, p. 673. *Oil Required for Lubricators:* Elec. World, May 5, 1906, p. 934. *Gumming Tests:* Jour. Am. Chem. Soc., April, 1902, p. 467. *Valuation of Lubricants:* Jour. Am. Chem. Ind., April 15, 1905, p. 315.

Lubrication, General: Power, March, 1903, p. 135; Mech. Engr., June 30, 1906, p. 919; Prac. Engr., Dec. 15, 1905, p. 915; Elec. Engr., Lond., Sept. 7, 1906, p. 344.

Oil Purification: Elec. Engr., Lond., Jan. 13, 1903, p. 51; Elec. World, Dec. 1, 1906, p. 1053.

Economy in Lubrication of Machinery: Trans. A.S.M.E., 4-315. *Theory of Finance of Lubrication:* Trans. A.S.M.E., 6-437.

Experiments, Formulas, and Constants for Lubrication of Bearings: Am. Mach., Sept. 10, 1903, pp. 1281, 1316, 1350.

Lubricators and Lubricants: Power & Engr., Sept. 21, 1909, p. 486.

Selection of an Oil for Lubrication: Power & Engr., July 27, 1909, p. 137.

CHAPTER XVII.

FINANCE AND ECONOMICS — COST OF POWER.

388. Records. — Few engineers realize the importance of a detailed system of accounting, or the saving which may be effected in cost of operation by careful study of the daily records of performance. Many regard graphical load curves, meter readings, and similar records as interesting but of little economic value. During the past few years the author has made a close study of the cost of power in a large number of central and isolated stations in Chicago, and found, without exception, that the highest economy was effected by the engineers who kept the most systematic records; the poorest results were obtained where records were kept indifferently or not at all. In some small plants the numerous duties of the engineer prevented him from devoting the necessary time, but in the majority of cases the absence of records was due entirely to lack of interest. Power-plant records to be of value must be closely studied with a view to improvement. The mere accumulation of data to be filed away and never again referred to is a waste of time and money.

Records should cover not only the daily operation of the plant but also, as permanent statistics, a complete analysis of each item of equipment. The value of such data cannot be overestimated. The engineer will frequently find it greatly to his interest to have available at a moment's notice the complete details of his engines, boilers, generators, and other machinery, especially when it is required to renew a broken or worn-out part.

389. Output. — The periodical output of a power plant may be expressed in terms of

(1) Steam plant.

Indicated or brake horse power.
Indicated or brake horse-power hours.

(2) Electric lighting plant.

Electrical horse power or kilowatts.
Electrical horse-power hours or kilowatt hours.
Lamp hours.

(3) Electric railway plant.

Electrical horse power or kilowatts.
Electrical horse-power hours or kilowatt hours.
Car miles.

When a plant is operating at practically constant load it is sufficiently accurate for most purposes to express the output in horse power or kilowatts per year. When the output fluctuates from day to day it is best expressed in horse-power hours or kilowatt hours, or by specifying the load factor along with the periodical output in horse power. For example, 1 horse power per year, 24 hours per day and 365 days per year, is equivalent to $365 \times 24 = 8760$ horse-power hours. If the full power is used throughout this time, it matters little whether the charge is based on horse power or horse-power hours; if, however, the power is used say only half the time, the yearly cost per horse power will remain unchanged but the cost per horse-power hour will be just double. As will be shown later the load factor (ratio of actual to rated load) exerts a marked influence on the cost of producing power, and for this reason the output is usually expressed as horse-power hours, kilowatt hours, lamp hours, or the like.

390. Load Factor. — The *yearly load factor* or simply *load factor*, as it is usually called, is the ratio of the *actual* yearly output to the *rated* yearly output measured on a 24-hour basis. Thus:

For a steam plant,

$$\text{Load factor} = \frac{\text{Yearly output, horse-power hours}}{\text{Rated horse power} \times 8760}. \tag{170}$$

For an electric station,

$$\text{Load factor} = \frac{\text{Yearly output, kilowatt hours}}{\text{Rated capacity, kilowatts,} \times 8760}, \tag{171}$$

(8760 = number of hours in one year.)

The *curve load factor* or *station load factor* is the ratio of the actual yearly output to the rated output, based upon the number of hours the plant is in actual operation. Thus for an electric station,

$$\text{Curve load factor} = \frac{\text{Yearly output, kilowatt hours}}{\text{Rated capacity} \times \text{hours plant is in operation}}. \tag{172}$$

In any plant the great desideratum is a high load factor with consequent greatest return on the investment. All the factors of expense included in the cost of power are then operating at maximum economy. High peak loads and low average loads necessitate large machines which are but little used and greatly increase the fixed charges.

In any system the total fixed charges per year are constant irrespective of the load factor, since interest, taxes, depreciation, insurance, and maintenance go on whether the plant is in operation or not. The total fixed charges for a specific case are illustrated in Fig. 417 by a

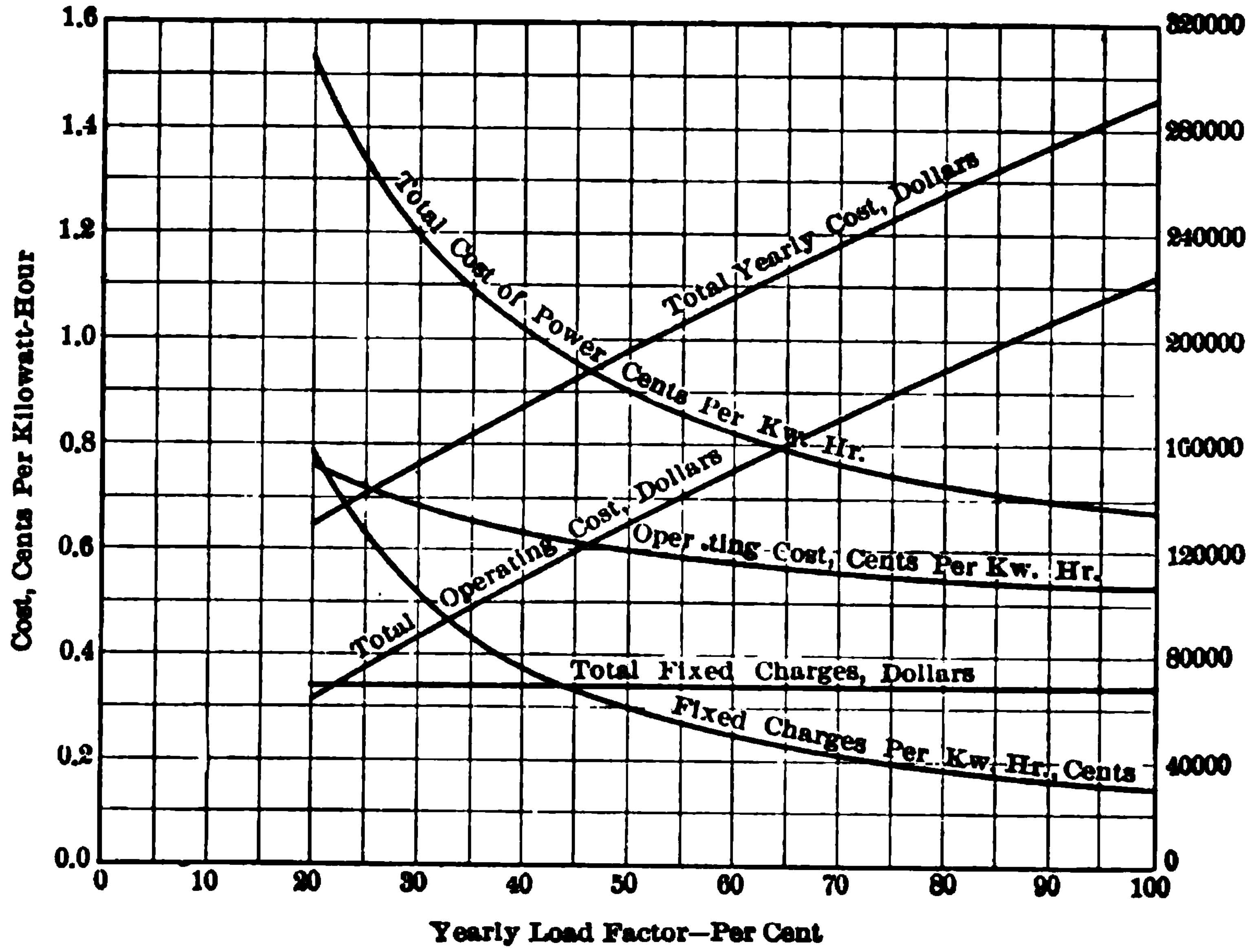


FIG. 417. Influence of Load Factor on the Cost of Power at the Switchboard. (5000 Kilowatt Electric Light and Power Station.)

straight line. The cost per kilowatt hour, however, decreases as the load factor increases. For example, with the plant operating continuously at rated load (100 per cent load factor) the fixed charges per kilowatt hour are

$$\frac{65{,}000}{5000 \times 8760} = \$0.00148.$$

With 30 per cent load factor these charges are

$$\frac{65{,}000}{0.3\ (5000 \times 8760)} = \$0.00445 \text{ kilowatt hour.}$$

The higher the load-factor the greater is the amount of power produced and the longer does the apparatus work at best efficiency. But the greater the power produced the larger will be the fuel consumption and the oil and supply requirements. The labor charges will be practically constant. The total operating cost per year increases as the load factor increases, but not directly. (See Fig. 417.) The cost per

kilowatt hour, however, decreases as the load factor increases. For example, the operating costs per year with plant operating continuously at full load are $230,200. This gives

$$\frac{230,200}{5000 \times 8760} = \$0.00525 \text{ per kilowatt hour.}$$

With 30 per cent load factor the yearly operating charges are $87,980, which gives

$$\frac{87,980}{0.3\,(5000 \times 8760)} = \$0.0067 \text{ per kilowatt hour.}$$

Table 107 shows the influence of the load factor on the cost of power in two isolated stations of the same rated capacity, one operating with the unusually high load factor of 80 per cent and the other operating with the low load factor of 17 per cent. The former furnishes current for a large electro-chemical concern in which the load is practically constant.

In general, the higher the load factor the greater becomes the ratio of the operating to the fixed charges, and extra investment may become advisable to secure the greatest economy possible.

On the other hand, when the load factor is low the fixed charges are the governing factor in the cost of power, and extra expenditures must be carefully considered, particularly if fuel is cheap.

391. Cost of Operation. — The cost of operation of power plants is conveniently divided into two parts:

(1) Fixed charges.
  (*a*) Investment costs.
  (*b*) Administration costs.
(2) Operating costs.

392. Fixed Charges. — These cover all expenses which do not expand and contract with the output. In very large plants they are usually divided into two parts, (*a*) the *investment costs*, which include interest, rental, depreciation, taxes, and insurance, and a reserve fund to cover depreciation of the investment, and (*b*) the *administration costs*, which include rental of offices, annual salaries of officers, and all other expenses not directly chargeable to the power plant. In the average plant the fixed charges comprise interest, rental, depreciation, taxes, insurance, and sometimes maintenance, though the latter is ordinarily included in the operating costs.

393. Interest. — The rates of interest on borrowed money vary with the nature of the security. In the case of power plants the form of security is usually a mortgage on the plant and equipment. If a

builder has sufficient funds to construct the plant without borrowing, he should charge against the item "interest" the income which the sum involved would bring if placed out at interest or if invested in his own business. In estimating the interest charges 5 per cent of the capital invested is ordinarily assumed unless specific figures are available.

TABLE 90.

APPROXIMATE USEFUL LIFE OF VARIOUS PORTIONS OF STEAM POWER PLANT EQUIPMENTS.

	Years.
Buildings, brick or concrete	50
Buildings, wooden or sheet iron	15
Chimneys, brick	50
Chimneys, self-sustaining steel	25
Chimneys, guyed sheet-iron	10
Boilers, water-tube	25
Boilers, fire-tube	15
Engines, slow-speed	25
Engines, high-speed	15
Turbines	25
Generators, direct-current	25
Generators, alternating-current	30
Motors	20
Pumps	25
Condensers, jet	35
Condensers, surface	20
Heaters, open	30
Heaters, closed	20
Economizers	20
Wiring	20
Belts	7
Coal conveyor, bucket	15
Coal conveyor, belt	10
Transformers, stationary	30
Rotary converters	25
Storage batteries	15
Piping, ordinary	12
Piping, first class	20

NOTE.— So much depends upon the design and the conditions of operation that no fixed values can be definitely assigned and the above figures should be used with caution. Practice shows that most power-plant appliances become obsolete long before the limit of their useful life is reached.

394. Depreciation. — This charge represents the gradual deterioration of a plant, resulting in its eventually wearing out. It is also assumed to represent the superannuation of a plant or the rate at which the apparatus is becoming obsolete. Thus, under the first assumption, if the useful life of an engine is 40 years, the rate of depreciation, neglecting interest, is 2.5 per cent; if, however, it is assumed that the engine will become obsolete in 20 years and uneconomical for further operation, the rate of depreciation will be 5 per cent. It is difficult to assign a fixed rate of depreciation against any

piece of apparatus, due to possible new developments which cannot be reckoned with in advance in computing the useful life of the apparatus. Again, depreciation cannot always be separated from current repairs and is a variable factor even in the parts of the same machine. It is therefore more or less of an approximation. The average life of various parts of a steam power plant is outlined in Table 90, but on account of the inability to assign fixed values to the useful life of any apparatus, and on account of the great number of appliances in even a small plant, it is customary to charge a fixed rate of depreciation against the entire plant and thus avoid confusion and complexity. This very crude method usually results in overestimation in well-designed, well-operated plants and underestimation in poorly designed and badly managed installations. One of the largest power plant designing concerns in Chicago charges 7½ per cent against depreciation and finds this figure none too small. The Pennsylvania Railroad uses 7 per cent to cover depreciation charges on their power-house equipments.

TABLE 91.

RATE OF DEPRECIATION.

(Per Cent of First Cost.)

Assumed Useful Life of Apparatus.	Rate of Interest, per Cent.												
	2	2.5	3	3.5	4	4.5	5	5.5	6	7	8	9	10
2	49.50	49.37	49.27	49.14	49.02	48.90	48.78	48.66	48.54	48.31	48.07	47.84	47.62
3	32.67	32.51	32.35	32.19	32.03	31.87	31.72	31.56	31.41	31.10	30.80	30.51	30.21
4	24.26	24.08	23.90	23.72	23.55	23.39	23.20	23.03	22.86	22.52	22.19	21.84	21.55
5	19.21	19.02	18.83	18.65	18.46	18.28	18.10	17.91	17.73	17.40	17.04	16.73	16.37
6	15.85	15.65	15.46	15.26	15.08	14.89	14.70	14.52	14.33	13.97	13.63	13.29	12.96
7	13.45	13.25	13.05	12.85	12.66	12.46	12.28	12.09	11.91	11.15	11.20	10.87	10.55
8	11.65	11.44	11.24	11.05	10.85	10.66	10.47	10.28	10.10	9.74	9.40	9.06	8.74
9	10.25	10.04	9.84	9.64	9.45	9.26	9.07	8.88	8.70	8.34	8.00	7.68	7.36
10	9.13	8.92	8.72	8.52	8.33	8.14	7.95	7.76	7.58	7.23	6.90	6.58	6.27
11	8.21	8.01	7.80	7.61	7.41	7.22	7.04	6.85	6.68	6.33	6.00	5.69	5.40
12	7.45	7.25	7.04	6.85	6.65	6.46	6.28	6.10	5.92	5.60	5.27	4.97	4.69
13	6.81	6.60	6.40	6.20	6.01	5.83	5.64	5.47	5.29	4.96	4.65	4.36	4.08
14	6.26	6.05	5.85	5.65	5.46	5.28	5.10	4.93	4.75	4.49	4.13	3.84	3.58
15	5.78	5.57	5.37	5.18	4.99	4.81	4.63	4.46	4.29	3.97	3.68	3.40	3.15
16	5.36	5.16	4.96	4.77	4.58	4.40	4.22	4.06	3.89	3.58	3.30	3.03	2.78
17	4.99	4.79	4.59	4.40	4.22	4.04	3.87	3.70	3.54	3.24	2.96	2.71	2.47
18	4.67	4.46	4.27	4.08	3.90	3.72	3.55	3.39	3.23	2.94	2.66	2.42	2.19
19	4.37	4.17	3.98	3.79	3.61	3.44	3.27	3.11	2.96	2.67	2.47	2.17	1.95
20	4.11	3.91	3.72	3.53	3.36	3.19	3.02	2.87	2.71	2.44	2.18	1.95	1.95
25	3.12	2.92	2.74	2.56	2.40	2.24	2.09	1.95	1.82	1.58	1.36	1.18	1.75
30	2.46	2.27	2.10	1.93	1.78	1.64	1.50	1.38	1.26	1.06	0.88	0.73	0.61
35	2.00	1.82	1.65	1.50	1.36	1.23	1.10	0.99	0.89	0.72	0.58	0.46	0.37
40	1.65	1.48	1.32	1.18	1.05	0.93	0.83	0.73	0.64	0.50	0.38	0.29	0.22
45	1.39	1.22	1.07	0.94	0.82	0.72	0.62	0.54	0.47	0.35	0.26	0.19	0.14
50	1.18	1.02	0.88	0.76	0.65	0.56	0.42	0.40	0.34	0.25	0.17	0.12	0.09

The rate of depreciation in terms of interest and useful life is a simple problem in compound interest, and may be expressed:

$$d = \frac{100\,r}{(1 + r)^n - 1}, \tag{173}$$

in which

d = rate of depreciation, per cent of first cost.

r = rate of interest.

n = assumed life of the apparatus in years.

This is based on the assumption that the interest is compounded annually and that the apparatus is valueless at the end of n years. Table 91 has been calculated with this formula and gives the rate of depreciation for different rates of interest and different asssumptions as to useful life of apparatus.

TABLE 92.

DEPRECIATION PERCENTAGES DETERMINED BY THE TRACTION VALUATION COMMISSION. POWER-PLANT DEPRECIATION.

Chicago, Ill., Sept. 8, 1906.	Per Cent.
Engines, Corliss, slow-speed	3 to 5
Engines, automatic, high-speed	5 to 10
Cable-winding machinery	3
Generators, direct connected, modern	5
Generators, belted (depending on date)	5 to 10
Traveling cranes	2
Switchboard and all wiring	2
Piping	35
Pumps	5
Heaters, closed	6 to 10
Heaters, open, if cast iron only	3
Breeching and connections, brick	5
Breeching and connections, steel	10
Boilers and settings, horizontal tubular	10
Boilers and settings, water-tube	3.5
Grates	10
Stokers	See below
Coal-handling machinery	6
Ash-handling machinery	8
Combined coal and ash-handling machinery	7
Storage bins, steel	3 to 10
Miscellaneous items	5

The above annual rates of depreciation have been used as a basis in depreciating the power-plant equipments. Apparatus has been depreciated at these rates down to 20 per cent of the wearing value, the wearing value being determined by subtracting the scrap value from the cost new. All power-plant equipment has been considered as worth 20 per cent of its wearing value as long as it is in operating condition. Depreciation applied to wearing value, as apparatus is always worth scrap value.

Stokers. The fixed parts depreciate very little and the moving parts and grates very rapidly, as the moving parts are renewed and maintained in good condition. All stokers in operation have been depreciated 25 per cent.

The above percentages applied to a particular plant of 3900 kilowatts capacity give an approximate depreciation for the whole plant of 4 per cent.

Table 92 gives the depreciation percentages determined by the Traction Valuation Commission, Chicago, Ill., as reported by the commission Sept. 18, 1906.

Example: A 1000-square-feet surface condenser and auxiliaries cost \$3500. With interest at 5 per cent, required the rate of depreciation, assuming a life of 25 years.

$$d = 100 \frac{0.05}{(1 + 0.05)^{25} - 1} = 2.09 \text{ per cent.}$$

That is to say, if 2.09 per cent of the first cost is laid aside each year for 25 years at 5 per cent interest, compounded annually, it will equal the first cost at the end of this period. The sum thus laid aside is sometimes called the *sinking fund.* The solution is more readily obtained with the aid of Table 91; e.g., at the intersection of vertical column 5 (interest) and horizontal column 25 (life in years) we find the depreciation 2.09 per cent. This sinking-fund method of rating the depreciation is peculiarly adapted to power-plant practice, inasmuch as a sum set aside at comparatively low rates of interest and compounded increases very slowly at first but more and more rapidly from year to year. This is precisely what happens in the deterioration of the plant. The loss of value is slight at first when the materials are new and usefulness is at a maximum, while towards the end of life both value and usefulness decline very rapidly.

If the apparatus still has some value at the end of n years and if b is the ratio of the value of the old material to that of the new, the rate of depreciation becomes

$$d = 100 \frac{r(1 - b)}{(1 + r)^n - 1}. \tag{174}$$

Example: In the foregoing problem, required the rate of depreciation if the value of the condenser outfit is \$350 at the end of 25 years.

$$\text{Here } b = \frac{350}{3500} = 0.1.$$

$$d = \frac{100 \times 0.05\,(1 - 0.1)}{(1 + .05)^{25} - 1}$$

$$= 1.97 \text{ per cent.}$$

That is, 1.97 per cent of \$3500, or \$68.95, laid aside each year for 25 years at 5 per cent interest and compounded annually will equal \$3500 − 350, or \$3150, at the end of this period.

It is not supposed that an owner will regularly lay aside this sum annually, or take the trouble to arrange for its investment at current rates in the market or savings bank, since the money is probably worth more to him in his own business. In practice it is retained in his business or investments and is earning the rate of interest obtainable therein, but in determining the net profit or loss this depreciation item is nevertheless accounted for just as if it were actually placed in outside investments.

In appraising the present value of any apparatus in terms of the rate of interest and useful life the expression becomes

$$V = 100\,\frac{(1+r)^m - 1}{(1+r)^n - 1}, \tag{175}$$

in which

V = total depreciation, per cent of original cost.
m = number of years apparatus has been in operation.
n = assumed life of apparatus.
r = rate of interest.

Example: In the preceding example, required the present valuation of the condenser, assuming that it has been in use 15 years.

$$m = 15,\ n = 25,\ r = 0.05.$$

Substituting these values in (175),

$$V = 100\,\frac{(1+0.05)^{15} - 1}{(1+0.05)^{25} - 1} = 45.1 \text{ per cent.}$$

That is, the condenser has depreciated 45.1 per cent of its original value and consequently is worth $\$3500 - 0.451 \times 3500 = \$1921.50$.

Table 91 may be conveniently used in this connection. At the intersection of vertical column 5 and horizontal columns 15 and 25 we find 4.63 and 2.09 respectively. Dividing 2.09 by 4.63 we get 0.451 = 45.1 per cent, the total depreciation. Depreciation is often taken care of under the different items pertaining to maintenance, and whenever a change or repair is necessary it is charged directly into expense as maintenance.

Though usually considered separately, interest and depreciation are sometimes considered as a single item, and in this case the rate of depreciation represents the sum which must be laid aside each year for the eventual renewal of apparatus plus the interest on the investment.

395. Maintenance. — Maintenance usually refers to the expense of keeping the plant in running order over and above the cost of attendance. It includes cost of upkeep, replacement, and precautionary measures. This latter item includes the renewal of working parts, painting of perishable or exposed material, and replacing worn-out and defective material. Many engineers make no allowance for maintenance in the fixed charges and include these costs under supplies, attendance, or repairs. In a general way, when maintenance is included under the fixed charges, an annual charge of 2 per cent is considered a liberal allowance, since most of the repair work comes under attendance. In street-railway practice maintenance is divided among the several parts of the system as follows: Buildings, steam appliances, electrical equipment, and miscellaneous. In this connection the maintenance becomes a part of the fixed charges, since the various items vary widely from month to month.

396. Taxes and Insurance. — Taxes vary from a fraction of one per cent to one and one-half per cent, depending upon the location of the plant. An average figure is one per cent of the actual value of the investment. Buildings and machinery are ordinarily insured against fire loss and boilers against accidental explosions, and accident policies are sometimes carried on all operating machinery. A fair charge for this item is one-half per cent.

397. Operating Costs. — Operating costs are conveniently divided as follows:

(1) Labor or attendance.
(2) Fuel and water.
(3) Oil, waste, and supplies.
(4) Repairs and maintenance.

In large stations it is often desirable to keep the expenses of the various departments separate from those of the power plant proper. Thus in central stations the distributing system is an important branch and the attending expenses form a considerable portion of the total. They are therefore kept separate, since they are not strictly chargeable to power generation. In isolated stations the wages of elevator men, though in a general way a part of the power-plant expenses, are not included in the "labor and attendance" charge of the plant. Lamps are a large item of expense in a tall office building, and for this reason are often kept separate from supplies.

398. Labor, Attendance, Wages. — The minimum number of men required to handle a given plant is approximately a fixed quantity and it is seldom possible to so arrange the work that any material

reduction can be effected. Until very recently it has been the universal custom to pay wages on a " flat rate " basis, that is, the attendant is given a fixed sum per day or month irrespective of the amount of work required or the economy of operation. In many cases, however, the bonus system has been successfully adopted. For example, in the boiler room the coal consumption is determined for a given period of time with ordinary careful firing, and the fireman is offered a reasonable percentage on the saving of coal which he is able to effect over this record by special care and attention to the keeping of fires always in the best condition, avoiding the blowing off of steam, using as little coal as needed for banking fires, and in other ways. Where careful records are kept of supplies, repairs, and renewals, the bonus is also applicable to electricians, oilers, and other employees.

Labor should always be estimated or recorded as so many dollars per month or per year and not merely in terms of the output unless the load factor is definitely known, otherwise comparisons are misleading. For example, consider two plants of 500 kilowatts capacity, each with labor charges, say, of $400 per month. Suppose the output of one is 100,000 kilowatt hours per month and that of the other 40,000 kilowatt hours per month. The monthly charges are evidently the same, viz., $400, but the cost per kilowatt hour differs widely, being 0.4 cent in the first case and 1 cent in the latter.

The cost of labor varies so much with the location of the plant and the conditions of operation that general figures are of little value except as a rough guide. The figures in Table 93 and Table 94 give average results for general practice. Specific figures will be found in Tables 95 to 107.

399. Cost of Fuel. — Tables 95 to 107 give examples of the cost of fuel in different types and sizes of steam power plants. It will be noted that this item varies considerably even with plants of the same general class. So much depends upon the market price of the fuel itself that the item " cost per horse-power or kilowatt hour " gives little information concerning the economy of operation unless the price of the fuel, its calorific value, and the water rate of the prime movers are specified. In a general sense the cost of fuel will range from 40 to 70 per cent of the total operating expenses. In estimating the cost of fuel for a proposed installation due consideration should be given to the coal burned in banking fires, heat lost in blowing off boilers, and reduced efficiency in operating at underloads and overloads. For example, individual tests of a number of boilers in a large central station in Chicago gave an average evaporation of 6.1 pounds of water per pound of coal, actual conditions, whereas the evaporation

TABLE 93.

COST OF LABOR FOR STREET-RAILWAY PLANTS.

(C. C. Moore & Co.)

Size of Plant, Kw.	Chief Engineer. No.	Chief Engineer. Rate per Month	Assistant Engineer. No.	Assistant Engineer. Rate per Month	Watch Engineer. No.	Watch Engineer. Rate per Month	Boiler-room Engineer. No.	Boiler-room Engineer. Rate per Month	Oilers. No.	Oilers. Rate per Month	Firemen. No.	Firemen. Rate per Month	Boiler Cleaners. No.	Boiler Cleaners. Rate per Month	Wipers. No.	Wipers. Rate per Month	Water Tenders. No.	Water Tenders. Rate per Month	Electricians. No.	Electricians. Rate per Month	Switchboard Men. No.	Switchboard Men. Rate per Month	Generator Men. No.	Generator Men. Rate per Month	Clerks. No.	Clerks. Rate per Month	Total Cost of Labor per Year.	Cost of Labor per Kw. per Year.	Cost of Labor in Dollars per Kw. Hour. .25	.33⅓	.50	.75	1.00
100	1	$85			1	$90					1	$60															$2,700	$27.00	.01233	.00925	.00616	.00410	.00308
200	1	85			1	80					1	60															2,700	13.50	.00616	.00463	.00308	.00205	.00154
250	1	85			1	80					1	60															2,700	10.80	.00493	.00370	.00247	.00164	.00123
300	1	90			1	80					1	60															2,700	9.20	.00420	.00315	.00210	.00140	.00105
400	1	90			1	80					1	60			1	60											3,450	8.70	.00397	.00298	.00199	.00132	.00099
500	1	90			1	85			1	60	1	60			1	60											4,260	8.52	.00389	.00292	.00195	.00130	.00097
750	1	90			1	85			1	60	2	70			1	60											5,220	6.96	.00318	.00238	.00159	.00106	.00079
1000	1	110			1	90			2	60	2	70			2	60											6,960	6.96	.00318	.00238	.00159	.00106	.00079
1500	1	120			1	90			2	60	2	70	1	65	2	60							2	60			9,300	6.20	.00283	.00212	.00142	.00094	.00071
2000	1	125			2	100			3	60	3	1–65 2–70	1	65	2	60							2	60			12,180	6.09	.00278	.00209	.00139	.00093	.00070
2500	1	125			2	100			4	60	3	1–65 2–70	1	70	2	60							2	70			13,200	5.28	.00241	.00181	.00121	.00090	.00060
3000	1	125			2	100			4	65	3	70	3	2–60 1–70	2	60							2	70			14,940	4.98	.00227	.00171	.00114	.00076	.00057
4000	1	125			2	100			4	65	3	75	3	2–65 1–70	2	60							2	70			15,240	3.81	.00174	.00130	.00087	.00058	.00043
5000	1	135			3	100			5	65	3	75	4	3–65 1–70	3	60					1	75	3	1–65 2–70			20,520	4.10	.00187	.00141	.00094	.00062	.00047
7500	1	135			3	110	1	90	8	65	5	75	6	65	3	60					2	80	3	65	1	70	29,340	3.91	.00179	.00134	.00089	.00060	.00045
10000	1	150	1	120	2	120	1	90	10	65	6	75	7	65	4	60	3	75			3	1–85 2–80	3	65	2	1–65 1–70	37,560	3.76	.00172	.00129	.00086	.00057	.00043
15000	1	200	1	125	5	110	1	100	15	65	10	75	10	65	6	60	3	75	1	90	3	85	3	65	2	1–65 1–70	55,320	3.69	.00168	.00126	.00084	.00056	.00042
20000	1	250	2	125	5	110	1	100	20	65	12	75	12	2–70 10–65	8	60	6	75	1	90	3	85	5	65	3	2–65 1–70	71,280	3.56	.00162	.00122	.00081	.00054	.00041

NOTE. — Above figures [illegible] ınits and conditions of service.

Yearly load fa[illegible]

All plants bet[illegible] current; all others alternating current.
All plants 4000 [illegible] continuously for 20 hours per day, 2–10 hour shifts; all others 24 hours per day, 3–10 hour shifts.

TABLE 94.

COST OF LABOR IN TALL OFFICE BUILDING POWER PLANTS. SERVICE 18 HOURS, 2-10 HOUR SHIFTS.

(C. C. Moore & Co.)

determined by dividing the total water fed into the boiler per year by the total consumption of coal gave only 5.2 pounds. Current practice gives an average efficiency (based on yearly operation) of boiler and furnace of 70 per cent for pumping stations running at practically full load, 65 per cent for large lighting and power stations with yearly load factor of 0.50 or more, and 60 per cent for similar stations with load factor between 0.35 and 0.45. For very low load factors, 0.25 and under (as in connection with manufacturing plants, tall office building and other plants operating on a 10-hour basis) this efficiency seldom exceeds 50 per cent. With these figures as a guide the cost of fuel per unit output may be roughly approximated.

400. Oil, Waste, and Supplies. — These items approximate 2 to 10 per cent of the total operating expenses. Tables 95 to 107 give some idea of current practice in different classes of power plants.

401. Repairs and Maintenance. — This item ordinarily refers to the cost of keeping the plant in running order over and above the cost of labor or attendance, and depends upon the age and condition of the plant and the efficiency of the employees. Tables 95 to 107 give the cost of repairs and maintenance for a wide range in power-plant practice.

402. Cost of Power. — The actual cost of producing power depends upon the geographical location of the plant, the size of apparatus, the design, conditions of loading, system of distribution, and the method of accounting. Tables 95 to 108 compiled from various sources give the detailed costs of a large number of central and isolated stations.

Table 95. Operating costs per kilowatt hour for a number of typical British electric light and power plants.

Table 96. Operating costs per kilowatt hour for a number of United States electric power plants for street-railway, light, and general power service.

Table 97. Average operating costs per kilowatt hour for all stations of the Boston Elevated.

Table 98. Operating costs for the year 1907 of the mechanical plant of the First National Bank Building, Chicago.

Table 99. Costs, fixed and operating, of producing one brake horse power per year, simple non-condensing engine, etc.

Table 100. Cost of power for compound condensing engine plants.

Table 101. Costs of different sizes and types of plants and annual costs per brake horse power, average working conditions.

Tables 102, 103, and 104. Cost, fixed and operating, of producing one electrical horse power per year for different sizes and types of plants.

Table 105. Influence of load factor on cost of electrical power in isolated stations.

Tables 106 and 107. Cost of furnishing heat, light, and power for a number of isolated stations in New York City, tall office buildings, loft buildings, apartment houses, hotels, and club buildings.

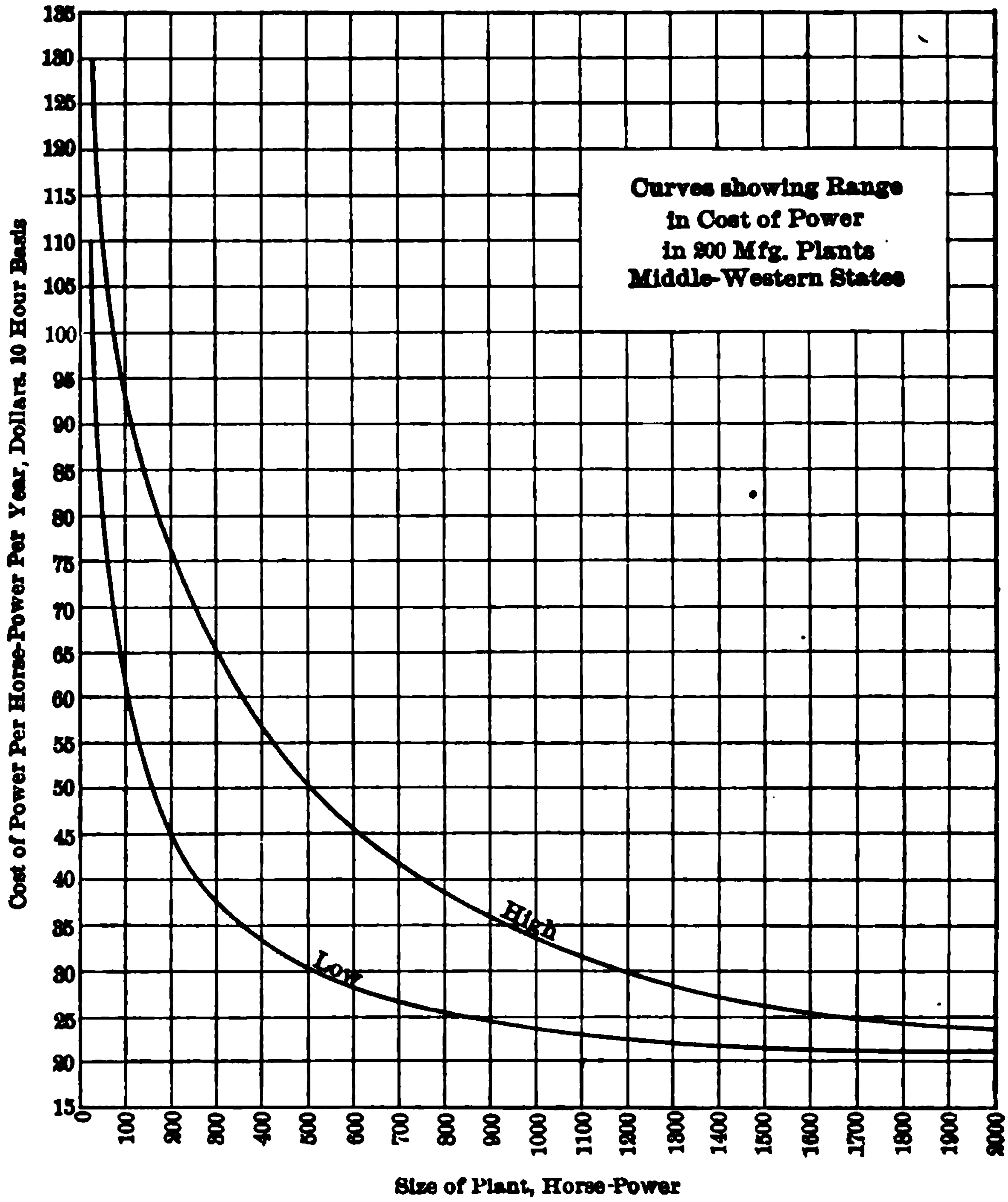


FIG. 418.

Table 108. Cost of furnishing heat, light, and power for the year 1907, First National Bank Building, Chicago.

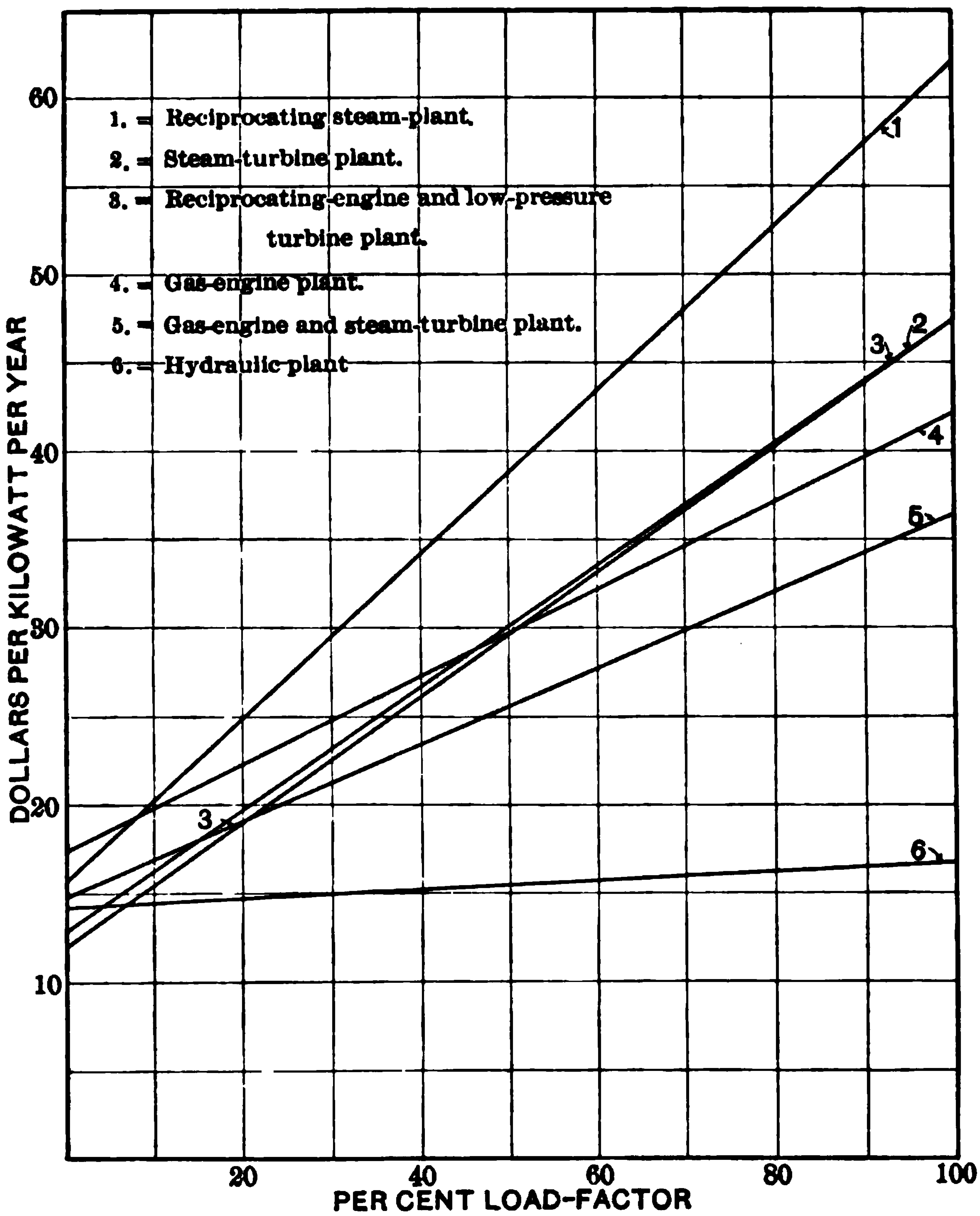


Fig. 418a. Cost of Power in Large Power Plants with Maximum Load over 30,000 Kilowatts. Coal at $3.00 per Ton. 14,500 B.T.U. per Pound.

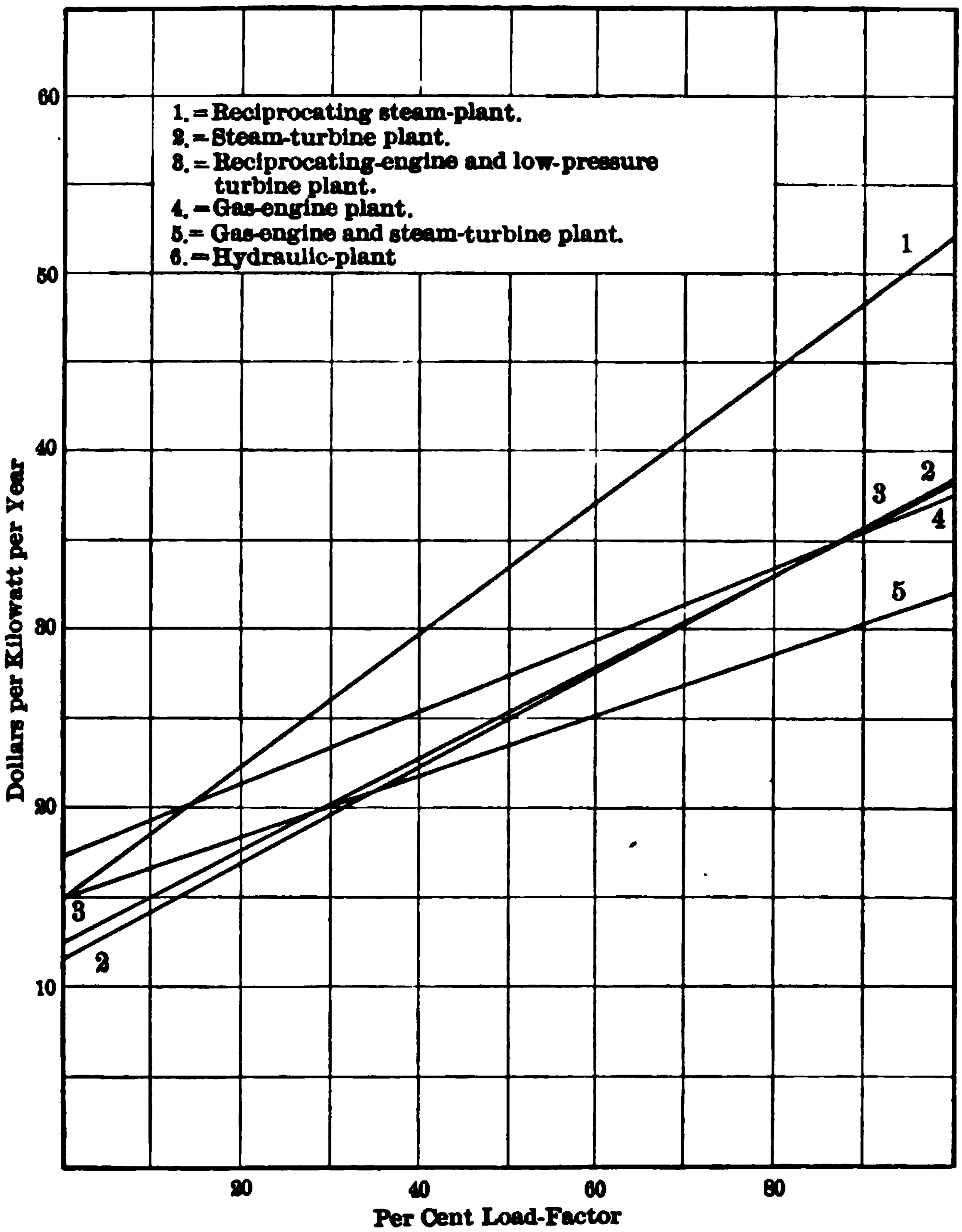


FIG. 418b. Cost of Power in Large Power Plants with Maximum Load over 30,000 kilowatts. Coal at $1.50 per Ton. 11,000 B.T.U. per Pound.

Table 94a, and Figs. 418a, 418b and 418c give the fundamental relations between the various items entering into the cost of power for various types of plants of over 30,000 kilowatts rated capacity. These data are taken from a paper presented by H. G. Stott at a meeting of the Toronto section of the American Institute of Electrical Engineers, Toronto, Ont., December, 1908. The figures have been brought up to date (June, 1910) by Mr. Stott and show what is actually being done to-day in large plants of the size stated above.

TABLE 94a.

DISTRIBUTION OF MAINTENANCE AND OPERATION COSTS IN POWER PLANTS HAVING A MAXIMUM LOAD OVER 30,000 KILOWATTS.

(H. G. Stott.)

	Reciprocating Steam Plant.	Steam Turbine Plant.	Reciprocating Engines and Low-pressure Steam Turbines.	Gas Engine Plant.	Gas Engines and Steam Turbines.	Hydraulic.
Maintenance.						
1. Engine room, mechanical	2.59	0.51	1.55	5.18	2.84	0.51
2. Boiler or producer room	4.65	4.33	3.55	1.16	1.97	......
3. Coal- and ash-handling apparatus	0.58	0.54	0.44	0.29	0.29	......
4. Electrical apparatus	1.13	1.13	1.13	1.13	1.13	1.13
Operation.						
5. Coal	61.70	55.53	46.48	26.52	25.97	......
6. Water	7.20	0.65	0.61	3.60	2.16	......
7. Engine room, labor	6.75	1.36	4.06	6.76	4.06	1.36
8. Boiler or producer room, labor	7.20	6.74	5.50	1.81	3.05	......
9. Coal- and ash-handling, labor	2.28	2.13	1.75	1.14	1.14	......
10. Ash removal	1.07	0.95	0.81	0.54	0.54	......
11. Electrical labor	2.54	2.54	2.54	2.54	2.54	2.54
12. Engine room, lubrication	1.78	0.35	1.02	1.80	1.07	0.20
13. Engine room, waste, etc	0.30	0.30	0.30	0.30	0.30	0.20
14. Boiler room, lubrication, etc.	0.17	0.17	0.17	0.17	0.17	......
Relative operating cost, per cent	100.00	77.23	69.91	52.94	47.23	5.94
Relative investment, per cent	100.00	75.00	80.00	110.00	96.20	100.00
Probable average cost per kilowatt	125.00	93.75	100.00	137.50	120.00	125.00
Probable fixed charges	11%	11%	11%	12%	11.5%	11%

For steam-turbine plants larger than 60,000 kw. the cost per kilowatt may be reduced to $75.00.

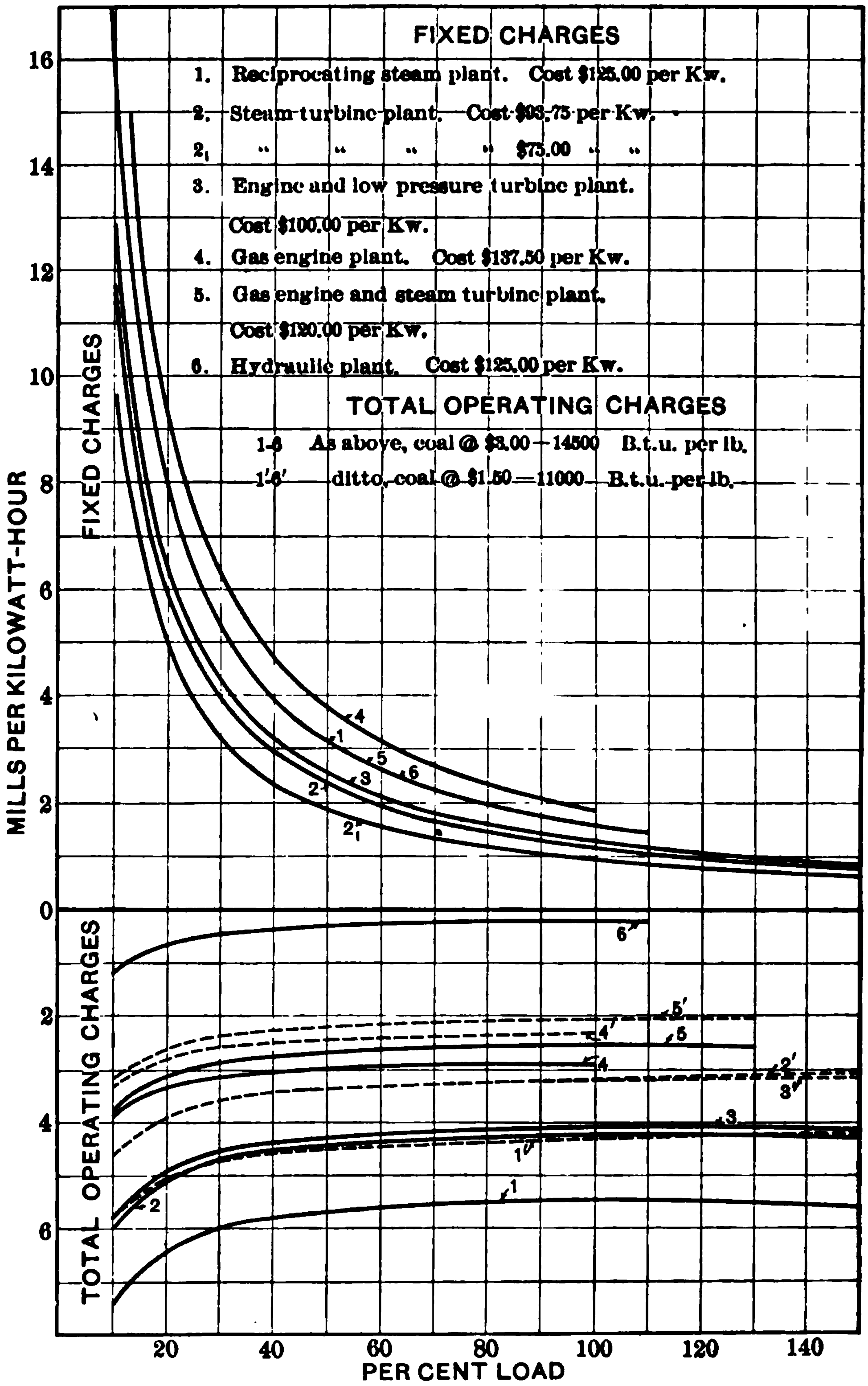


Fig. 418c. Cost of Power in Large Power Plant with Maximum Load over 30,000 kilowatts.

TABLE 95.

OPERATING COSTS, CENTS PER KILOWATT HOUR. TYPICAL BRITISH ELECTRIC LIGHT AND POWER PLANTS, 1902.

(*Engineering Record*, March 26, 1904, p. 389.)

Locality.	Kilowatts Installed.	Yearly Load Factor.	Coal.	Oil, Waste, and Supplies.	Wages.	Repairs.	Total.
		Per Cent.					
Bradford	6,380	20.93	0.52	0.10	0.16	0.26	1.04
Leeds	8,740	12.31	0.56	0.06	0.34	0.28	1.24
St. Helen's	1,340	17.84	0.52	0.06	0.34	0.38	1.3
Edinburgh	10,477	14.75	0.68	0.08	0.18	0.36	1.3
Bolton	3,700	18.87	0.70	0.12	0.3	0.20	1.32
Booth	850	28.44	0.82	0.06	0.3	0.22	1.4
Liverpool	21,190	25.11	0.74	0.12	0.3	0.26	1.42
South Shields	1,600	15.82	0.74	0.08	0.4	0.30	1.52
Nottingham	5,642	12.97	0.92	0.20	0.32	0.18	1.62
Preston	1,920	13.31	0.72	0.12	0.36	0.46	1.66
Farnsworth	610	14.54	0.92	0.20	0.36	0.22	1.7
Leith	990	19.79	1.10	0.08	0.42	0.18	1.78

TABLE 96.

OPERATING COSTS, CENTS PER KILOWATT HOUR. TYPICAL STREET-RAILWAY PLANTS, UNITED STATES.

(*Street Railway Review*, Oct. 20, 1902, p. 774.)

Items.	1	2	3	4		5
	Two Stations Supplying Power for 1000 Cars, 175 Miles of Track. (Belted Installation.)	One Station, Power and Light, 500 Cars, 230 Miles of Track.	A Large City Central Station.	Combined Light and Power Station.		Inter-urban, Light and Power.
				July.	November.	
Fuel	0.68	0.803	0.509	0.958	0.870	1.004
Wages	0.199	0.133	0.115	0.287	0.276	0.217
Oil and waste	0.013	0.018	0.012	0.048	0.025	0.023
Water	0.04	....	.009	....	....	0.022
Maintenance:						
Buildings	0.005	....	....	....	....	0.015
Steam appliances	0.072	0.029	.011	....	....	0.035
Electric equipment	0.008	0.019	.001	0.004	....	0.010
Miscellaneous	0.033	0.023	.012	....	....	....
Total	1.05	1.03	0.669	1.299	1.188	1.326
Cost of coal, dollars per ton	1.55	2.66	3.07	2.98	2.98	3.62
Output, kilowatt hours	20,981,295	2,140,641	641,650	119,304	114,384	2,104,337
Coal per kilowatt hour, pounds	8.8	6.04	....	6.4	5.55	5.55

TABLE 97.

OPERATING EXPENSES OF BOSTON ELEVATED RAILWAY COMPANY.
AVERAGE OF ALL STATIONS AT SWITCHBOARD, APRIL 8, 1910.

	Year.							
	1902	1903	1904	1905	1906	1907	1908	1909
Rated capacity, kilowatts	35544	35544	35544	35544	38469	39969	50425	50063
Yearly load factor	.386	.391	.417	.437	.43	.45	.37	.37
Coal.* Cents per K.W.H.	0.47	0.78	0.53	0.45	0.47	0.55	0.56	0.45
Labor. " " "	0.19	0.18	0.16	0.15	0.17	0.18	0.21	0.20
Repairs.† " " "								
Supplies " " "	0.58	0.94	0.67	0.57	0.60	0.76	0.86	0.61
Total " "	0.77	1.12	0.83	0.72	0.77	0.94	1.07	0.81
Ratio operating expense to gross earnings	0.694	0.690	0.696	0.68	0.687	0.69	0.67	0.654
Price of coal per ton	$3.60	$4.854	$3.55	$3.1354	$3.1859	$3.572	$3.568	$3.209

* Coal included in supplies. † Repairs included in supplies and labor.

TABLE 98.

COST OF OPERATION (1907). FIRST NATIONAL BANK BUILDING, CHICAGO.

Coal bill	$34,567.27
Ash cartage	2,075.83
Water bill	3,535.75
Electrical supplies and repairs	4,016.68
Engine-room supplies and repairs	1,311.36
Boiler-room supplies and repairs	1,316.50
Oil, waste, and grease	1,554.80
Packings	727.01
Machine-shop supplies	187.13
Refrigerating supplies	761.81
Steam fitting	499.53
Steam heat supplies	110.20
Plumbing supplies	184.87
Lamps	1,655.22
Wages	24,572.89
Petty expenses	24.62
Office expenses	186.55
Doctor bills	43.00
Coal analysis	160.00
Total	$77,591.02
Total receipts from all sources for power, heat, light, and miscellany	71,435.85
Net cost	$6,055.17

TABLE 99.

COST OF ONE HORSE POWER PER YEAR, SIMPLE ENGINES, NON-CONDENSING, 10-HOUR BASIS, 308 DAYS PER YEAR.

(Wm. O. Webber, *Engineer* U.S., March 16, 1903, p. 241.)

Size of plant................horse power	20	40	60	80
Cost of plant per horse power...........	$200.00	$190.00	$180.00	$175.00
Fixed charges at 14 per cent	28.00	26.60	25.20	24.50
Coal per horse-power hour, in pounds....	12.00	10.00	9.00	8.00
Cost at $4.00 per ton	66.00	55.00	49.50	44.00
Attendance, 10-hour basis..............	30.00	20.00	15.00	13.00
Oil, waste, and supplies.................	6.00	4.00	3.00	2.60
With coal at $5.00 per ton...............	146.50	119.35	105.07	95.10
With coal at $4.50 per ton...............	138.25	112.47	98.80	89.60
With coal at $4.00 per ton...............	130.00	105.60	92.70	84.10
With coal at $3.50 per ton...............	121.75	98.72	86.51	78.60
With coal at $3.00 per ton...............	113.50	91.85	80.32	73.10
With coal at $2.50 per ton...............	105.25	84.97	74.13	67.60
With coal at $2.00 per ton...............	97.00	78.10	67.95	62.10

TABLE 100.

COST OF ONE HORSE POWER PER YEAR, COMPOUND CONDENSING ENGINES, 10-HOUR BASIS, 308 DAYS PER YEAR.

(Wm. O. Webber, *Engineer* U.S., Feb. 2, 1903, p. 144.)

Size of plant.........horse power	100	200	300	400	500	600
Cost of plant per horse power....	$170.00	$146.00	$126.00	$110.00	$96.00	$85.00
Fixed charges at 14 per cent.....	23.80	24.40	17.65	15.40	13.45	11.90
Coal per horse-power hour, pounds	7.0	6.5	6.0	5.5	5.0	4.5
Cost of fuel at $4.00 per ton......	38.50	35.70	33.00	32.00	27.50	24.70
Attendance, 10-hour basis........	12.00	10.00	8.60	7.25	6.20	5.40
Oil, waste, supplies.............	2.40	2.00	1.72	1.45	1.24	1.08
Total	76.70	68.10	60.97	56.10	48.39	43.08
With coal at $5.00 per ton........	86.40	77.10	69.22	61.90	55.29	49.28
With coal at $4.50 per ton........	81.50	72.60	65.07	58.10	51.79	46.18
With coal at $4.00 per ton........	76.70	68.10	60.97	56.10	48.39	43.08
With coal at $3.50 per ton........	71.90	63.70	56.82	50.50	45.04	39.98
With coal at $3.00 per ton........	67.00	59.20	51.67	46.70	41.49	36.88
With coal at $2.50 per ton........	62.30	54.75	48.59	43.00	38.83	33.83
With coal at $2.00 per ton........	57.45	50.25	44.47	40.10	34.64	30.73

Size of plant.........horse power	700	800	900	1000	1500	2000
Cost of plant per horse power....	$76.00	$69.00	$64.00	$60.00	$58.00	$56.00
Fixed charges at 14 per cent.....	10.65	9.65	8.95	8.40	8.12	7.85
Coal per horse-power hour, pounds	4.0	3.5	3.0	2.5	2.0	1.5
Cost of fuel at $4.00 per ton......	22.00	19.20	16.50	13.75	11.00	8.25
Attendance, 10-hour basis........	4.70	4.15	3.75	3.50	3.25	3.00
Oil, waste, supplies.............	0.94	0.83	0.75	0.70	0.65	0.60
Total	38.29	33.83	29.95	26.35	23.02	19.70
With coal at $5.00 per ton........	43.79	39.73	34.05	29.80	25.77	21.75
With coal at $4.50 per ton........	41.04	36.28	32.00	28.05	24.39	20.72
With coal at $4.00 per ton........	38.29	33.83	29.95	26.35	23.02	19.70
With coal at $3.50 per ton........	35.54	31.48	27.87	24.60	21.64	18.67
With coal at $3.00 per ton........	32.79	29.03	25.80	22.90	20.27	17.65
With coal at $2.50 per ton........	30.04	27.18	23.75	21.20	18.89	16.60
With coal at $2.00 per ton........	27.29	24.23	21.70	19.47	17.52	15.57

TABLE 101.

SHOWING CAPITAL COSTS OF PLANTS INSTALLED AND ANNUAL COSTS OF POWER PER BRAKE HORSE POWER, AVERAGE WORKING CONDITIONS.

(H. Von Schon, *Engineering Magazine*, May, 1907.)

Class I. — Engines: Simple, Slide-Valve, Non-Condensing.
Boilers: Return Tubular.

Size of Plant, Horse Power.	Engines, Boilers, etc., Installed.	Capital Cost of Plant per Horse Power Installed.		Annual Cost of 10 Hours Power per B.H.P.	Annual Cost of 20 Hours Power per B.H.P.
		Building.	Total.		
10	$66.00	$40.00	$106.00	$91.16	$180.76
20	56.00	37.00	93.00	76.31	151.48
30	48.70	35.00	83.70	66.46	131.68
40	44.75	33.50	78.25	59.49	117.74
50	43.00	31.00	74.00	53.95	106.46

Class II. — Engines: Simple, Corliss, Non-Condensing.
Boilers: Return Tubular.

30	$70.70	$35.00	$105.70	$61.14	$117.70
40	62.85	33.50	96.35	55.50	107.10
50	59.00	31.00	90.00	50.70	97.73
60	56.00	30.00	86.70	47.42	91.34
80	50.00	27.50	77.50	43.86	85.41
100	44.60	25.00	69.60	40.55	79.19

Class III. — Engines: Compound, Corliss, Condensing:
Boilers: Return Tubular with Reserve Capacity.

100	$63.40	$28.00	$91.40	$33.18	$60.05
150	53.70	24.00	77.70	29.83	54.63
200	50.10	20.00	70.10	28.14	51.72
300	45.90	18.00	63.90	26.27	48.83
400	43.55	16.00	59.55	24.84	46.12
500	41.25	14.00	55.25	23.73	44.21
750	40.50	13.00	53.50	23.56	44.02
1000	39.00	12.00	51.00	23.26	43.71

TABLE 102.

COST OF ELECTRICAL POWER (W. M. WILSON).

(*Power*, January, 1907, p. 15.)

Capacity of Plant I.H.P.	Type of Engine	Steam Pressure-Gauge	Lb. of Steam per 1 H.P.-H.	Size of Engines	No. of Engines	Cost of Engines	Cost of Generators	Cost of Pipe Steam	Cost of Condensers	Cost of Engine Room	Boiler H.P. Required	Type of Boilers	No. of Boilers	Size of Boilers	Cost of Boilers	Cost of Boiler Room	Cost of Feed Pumps	Cost of Economizers	Cost of Heaters
100	Simple Non-Cond. H. S.	100	30	100	2	2880	2300	1000		1292	92	F.T.	2	92	1904	2680	280		180
	Compound Non-Cond. H. S.	120	24	100	2	4600	2300	1000		1480	74	F.T.	2	74	1640	2510	230		180
	Compound Cond. H. S.	120	20	100	2	4600	2300	1000	1400	1430	66	F.T.	2	66	1500	2414	200		125
	De Laval Turbine	120	18.6	100	2	7400		1000	1200	910	62	F.T.	2	62	1425	2360	200		120
200	Simple Non-Cond. H. S.	100	30	100	3	4290	3450	1500		1830	186	F.T.	3	93	2850	3850	530		235
	Compound Non-Cond. H. S.	120	24	100	3	6300	3450	1500		2040	148	F.T.	3	74	2460	3600	440		200
	Compound Cond. H. S.	120	20	100	3	6900	3450	1500	2050	2040	132	F.T.	3	66	2250	3440	400		180
	De Laval Turbine	120	18.6	100	3	11100		1500	1940	1270	124	F.T.	2	124	2480	2825	390		180
400	Simple Non-Cond. H. S.	100	30	200	3	7020	6630	3000		2450	370	F.T.	4	123	4920	6312	1000		400
	Compound Non-Cond. H. S.	120	24	200	3	10620	6630	3000		2898	297	F.T.	4	99	3960	5098	840		336
	Compound Cond. H. S.	120	20	200	3	10620	6630	3000	3700	2700	264	F.T.	4	88	3670	4970	740		312
	De Laval Turbine	120	17	200	3	18960		3000	3250	1620	226	F.T.	3	113	3390	4000	650		275
	Vertical Cond. L. S.	150	13	400	2	20400	10600	4000	2600	4300	153	W.T.	2	153	6100	2300	500	665	200
	Hor. Cond. L. S.	150	13	400	2	13600	15400	4000	2600	5440	153	W.T.	2	153	6100	2300	500	665	200
600	Simple Non-Cond. H. S.	100	30	200	4	9360	8840	4000		3190	555	F.T.	6	111	6660	7670	1450		560
	Compound Non-Cond. H. S.	120	24	200	4	14160	8840	4000		3512	445	F.T.	5	111	5550	6440	1200		400
	Compound Cond. H. S.	120	20	200	4	14160	8840	4000	5060	3512	397	F.T.	5	99	4950	6298	1070		420
	De Laval Turbine	120	17	200	4	25280		4000	4530	2094	339	F.T.	4	113	4520	5240	930		380
	Vertical Cond. L. S.	150	13	300	3	29550	12600	4800	3600	5225	230	W.T.	2	230	8460	2530	650	1150	280
	Hor. Cond. L. S.	150	13	300	3	19800	19920	4800	3600	7020	230	W.T.	2	230	8460	2530	650	1150	280
	Parsons Turbine	150	13	600	2	35000		6000	3600	1470	230	W.T.	2	230	8460	2530	650	1150	280
1200	Simple Non-Cond. H. S.	100	30	400	4	15200	18000	8000		4030	1110	F.T.	10	122	12300	12776	2500		1060
	Compound Non-Cond. H. S.	120	24	400	4	22280	18000	8000		4840	890	F.T.	8	127	10160	10340	2120		860
	Compound Cond. H. S.	120	20	400	4	22280	18000	8000	8100	4640	794	F.T.	8	113	9040	10128	1950		780
	De Laval Turbine	120	16.6	800	5	45030		7500	7200	2960	660	F.T.	7	110	7700	8880	1700		660
	Vertical Cond. L. S.	150	13	600	3	43200	21800	9000	6100	7132	460	W.T.	3	230	12690	3660	1220	2300	480
	Hor. Cond. L. S.	150	13	600	3	28800	28020	9000	6160	8420	400	W.T.	3	230	12690	3660	1220	2300	480
	Parsons Turbine	150	13	1200	2	46000		12000	6100	2800	460	W.T.	3	230	12690	3600	1220	2300	460
2000	Vertical Cond. L. S.	150	13	500	5	60000	81350	12500	8450	10735	768	W.T.	4	250	18400	4900	1520	3840	700
	Hor. Cond. L. S.	150	13	1000	3	48000	34650	15000	8450	9320	768	W.T.	4	256	18400	4900	1920	3340	700
	Parsons Turbine	150	13	1000	3	62400		15000	8450	3360	766	W.T.	4	256	18400	4900	1520	3840	700

TABLE 102 (*Continued*).

TABLE 103.

COST OF ONE HORSE POWER PER YEAR IN STREET-RAILROAD SERVICE FOR DIFFERENT CLASSES OF ENGINES.

1000-Horse-Power Plant. (R. C. Carpenter.)

(Sibley, *Journal of Engineering*, November, 1904, p. 92.)

	Coal per Horse Power, Pounds per Hour.	Tons per Horse Power per Year. Day—12 Hours.	Day—18 Hours.	Day—24 Hours.	Fuel Cost per Year, 18 Hours per Day, $2.00 per Ton.	Oil, Waste, and Labor.	Interest, 5 per Cent; Depreciation, 10 per Cent.	Total Yearly Cost per Horse Power.
Non-Condensing Engines.								
Simple slide-valve, average	4.63	10.14	15.21	20.28	30.42	7.30	4.44	42.16
Simple slide-valve, best	4.60	10.07	15.10	20.14	30.20	7.30	4.44	41.94
Simple Corliss, average	3.45	7.55	11.33	15.10	22.66	7.30	4.75	34.71
Simple Corliss, best	3.01	6.59	9.89	13.18	19.78	7.30	4.75	31.83
*Compound slide-valve	4.17	9.05	13.57	18.10	27.14	6.90	4.80	38.84
Condensing Engines.								
Compound slide-valve, average	3.25	7.12	10.68	14.24	21.36	6.70	4.72	32.78
Compound slide-valve, best	2.40	5.25	7.88	10.51	15.76	6.50	4.72	26.98
Compound Corliss, average	2.36	5.17	7.74	10.33	15.48	6.50	5.28	27.26
Compound Corliss, best	1.80	3.94	5.91	7.88	11.82	6.10	5.28	23.20

* The compound slide-valve engine, running non-condensing, made, in this series of tests, a poorer record than the single Corliss. This may have been due to the extremely bad conditions of loading. It is, I think, a fact that this class of engine has not been a marked success for street railway work.

TABLE 104.

COST OF ELECTRICAL POWER — OIL FUEL.

Charles C. Moore & Co. Costs Based on Installation at Pacific Coast Railroad Terminal Points. San Francisco, Cal.

Total Rated Capacity of Plant — Kw.	Type of Engine.	No. of Units.	Size of Units — Kw.	Total First Cost of Plant — Dollars. Does not Include Real Estate or Piping Outside of Building Walls.	Total Fixed Charges per Year — Dollars.	Total Cost of Labor per Year — Dollars.	Kw. Hours per Barrel of Oil 12-Hour Test.	100 per Cent Curve Load Factor. Supplies per Kw. Hour — Cents.	Water per Kw. Hour — Cents.	Kw. Hours per Barrel of Oil.	Fixed Charges, Labor, Supplies, and Water per Kw. Hour — Cents.	Fuel per Kw. Hour at $1.00 per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel 50 Cents per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel 75 Cents per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel $1.00 per Barrel — Cents.
	24 Hours per Day Plants.														
50	High-speed tandem compound, non-condensing, D. C.	1	50	11,900	1,547	2,700	98	.060	.176	89	1.206	1.124	1.767	2.048	2.329
75	...do	1	75	15,200	1,976	2,700	101	.050	.176	92	.938	1.087	1.481	1.753	2.025
100	...do	1	100	18,100	2,353	2,700	110	.040	.169	100	.786	1.000	1.286	1.536	1.786
100	High-speed tandem compound, condensing, D. C.	1	100	20,300	2,639	2,700	116	.040	.016	105	.666	.952	1.142	1.380	1.618
150	...do	1	150	25,200	3,276	2,700	116	.035	.016	105	.506	.952	.982	1.220	1.458
300	Cross compound condensing, direct connected	1	300	44,200	5,746	3,840	175	.030	.011	159	.406	.629	.720	.877	1.035
500	...do	1	500	64,000	8,320	7,140	185	.020	.010	168	.383	.595	.681	.829	.978
750	...do	1	750	76,900	9,997	8,040	194	.020	.010	176	.305	.568	.589	.731	.873
1,000	...do	1	1,000	95,000	12,350	10,500	205	.017	.010	186	.288	.538	.557	.691	.826
1,500	...do	1	1,500	135,800	17,654	12,060	217	.017	.009	197	.252	.508	.506	.633	.760
2,500	...do	2	1,250	260,000	33,800	14,340	217	.017	.009	197	.246	.508	.500	.627	.753
5,000	...do	3	1,650	471,000	61,230	21,600	217	.015	.009	197	.213	.508	.467	.594	.721
10,000	...do	4	2,500	862,000	112,060	37,620	221	.013	.009	201	.193	.498	.442	.566	.690
20,000	Manhattan type, 4-cylinder, condensing, D. C.	4	5,000	1,480,000	192,400	71,280	225	.013	.009	205	.173	.488	.416	.538	.660

TABLE 104. — *Continued.*

20 Hours per Day Plants.

50	High-speed tandem compound, non-condensing, D. C.	1	50	11,900	1,547	2,700	98	.060	.176	86	1.400	1.136	1.968	2.282	2.636
75	do	1	75	15,200	1,976	2,700	101	.060	.176	91	1.080	1.039	1.620	1.904	2.179
100	do	1	100	18,100	2,353	2,700	110	.040	.169	99	.901	1.010	1.406	1.659	1.911
100	High-speed tandem compound, condensing, D. C.	1	100	20,200	2,630	2,700	116	.040	.016	106	.737	.962	1.264	1.502	1.740
150	do	1	150	25,200	3,276	2,700	116	.035	.016	105	.597	.953	1.073	1.311	1.549
300	Cross compound, condensing, direct connected	1	300	44,200	5,746	2,700	175	.030	.011	158	.429	.633	.746	.904	1.062
500	do	1	500	64,000	8,320	4,260	185	.020	.010	167	.375	.599	.674	.834	.973
750	do	1	750	76,900	9,997	5,220	194	.020	.010	175	.306	.571	.594	.737	.879
1,000	do	1	1,000	95,000	12,350	6,960	206	.017	.010	186	.292	.541	.562	.697	.832
1,500	do	1	1,500	135,800	17,654	9,300	217	.017	.009	195	.272	.513	.529	.657	.785
2,500	do	2	1,250	260,000	33,800	13,200	217	.017	.009	195	.264	.513	.540	.668	.796
5,000	do	3	1,660	471,000	61,230	20,520	217	.015	.009	195	.248	.513	.504	.633	.761
10,000	do	4	2,500	862,000	112,060	37,500	221	.013	.009	199	.237	.503	.478	.604	.730
20,000	Manhattan type, 4-cylinder, condensing, D. C.	4	5,000	1,490,000	193,400	71,280	225	.013	.009	203	.266	.493	.440	.572	.696

10 Hours per Day Plants.

50	High-speed tandem compound, non-condensing, D. C.	1	50	11,900	1,547	1,800	98	.060	.176	82	2.070	1.220	2.690	2.965	3.299
75	do	1	75	15,200	1,976	1,800	101	.060	.176	84	1.606	1.191	2.201	2.496	2.796
100	do	1	100	18,100	2,353	1,800	110	.040	.169	92	1.347	1.057	1.890	2.162	2.434
100	High-speed tandem compound, condensing, D. C.	1	100	20,300	2,630	1,800	116	.040	.016	97	1.272	1.081	1.788	2.045	2.309
150	do	1	150	25,200	3,276	1,800	116	.035	.016	97	.976	1.081	1.494	1.751	2.009
300	Cross compound, condensing, direct connected	1	300	44,200	5,746	4,740	175	.030	.011	146	.909	.695	1.341	1.512	1.684
500	do	1	500	64,000	8,320	4,740	185	.020	.010	154	.746	.649	1.070	1.233	1.396
750	do	1	750	76,900	9,997	5,560	194	.020	.010	162	.609	.617	.908	1.062	1.216
1,000	do	1	1,000	95,000	12,350	5,560	205	.017	.010	171	.518	.585	.811	.957	1.103
1,500	do	1	1,500	135,800	17,654	7,860	217	.017	.009	161	.492	.563	.768	.906	1.045
2,500	do	2	1,250	260,000	33,800	7,860	217	.017	.009	161	.463	.553	.759	.897	1.035

and are subject to considerable variation, depending on operating conditions, etc.; first cost depending on

perheaters.

at 150 pounds gauge pressure at boiler.

In all cases the circulating pumps are assumed to do 1,000 foot-pounds of work per pound of steam on main pounds of water against a 20-foot head. All plants located at tide water. Cooling towers increase first cost

TABLE 104.—(*Continued*).

Total Rated Capacity of Plant—Kw.	Type of Engine.	50 per Cent Curve Load Factor.						33⅓ per Cent Curve Load Factor.					
		Kw. Hours per Barrel of Oil.	Fixed Charges, Labor, Supplies, and Water per Kw. Hour—Cents.	Fuel per Kw. Hour at $1.00 per Barrel—Cents.	Total Expenses per Kw. Hour with Fuel 50 Cents per Barrel—Cents.	Total Expenses per Kw. Hour with Fuel 75 Cents per Barrel—Cents.	Total Expenses per Kw. Hour with Fuel $1.00 per Barrel—Cents.	Kw. Hours per Barrel of Oil.	Fixed Charges, Labor, Supplies, and Water per Kw. Hour—Cents.	Fuel per Kw. Hour at $1.00 per Barrel—Cents.	Total Expenses per Kw. Hour with Fuel 50 Cents per Barrel—Cents.	Total Expenses per Kw. Hour with Fuel 75 Cents per Barrel—Cents.	Total Expenses per Kw. Hour with Fuel $1.00 per Barrel—Cents.
	24 Hours per Day Plants.												
50	High-speed tandem compound, non-condensing, D. C.	70	2.270	1.429	2.984	3.341	3.698	56	3.320	1.786	4.212	4.659	5.105
75	...do.	72	1.740	1.389	2.434	2.782	3.129	58	2.528	1.724	3.391	3.822	4.253
100	...do.	79	1.446	1.266	2.079	2.396	2.712	63	2.094	1.587	2.888	3.285	3.682
100	High-speed tandem compound, condensing, D. C.	83	1.297	1.205	1.900	2.201	2.502	67	1.926	1.493	2.672	3.045	3.418
150	...do.	83	.981	1.205	1.583	1.188	2.186	67	1.453	1.493	2.199	2.572	2.946
300	Cross compound, condensing, direct connected	125	.787	.800	1.187	1.387	1.587	101	1.166	.990	1.661	1.908	2.156
500	...do.	132	.748	.758	1.127	1.316	1.506	106	1.111	.943	1.583	1.819	2.055
750	...do.	139	.591	.719	.951	1.131	1.310	111	.876	.901	1.326	1.552	1.777
1,000	...do.	146	.560	.685	.902	1.073	1.244	118	.830	.848	1.253	1.465	1.677
1,500	...do.	155	.489	.645	.811	.973	1.134	125	.724	.800	1.124	1.324	1.524
2,500	...do.	164	.474	.610	.779	.931	1.084	154	.696	.649	1.021	1.183	1.346
5,000	...do.	172	.408	.581	.699	.844	.990	166	.599	.602	.900	1.051	1.201
10,000	...do.	178	.369	.562	.650	.790	.931	169	.541	.592	.837	.985	1.138
20,000	Manhattan type, 4-cylinder, condensing, D. C.	181	.328	.553	.605	.743	.881	173	.480	.578	.769	.914	1.058

TABLE 104. — *Continued.*

20 Hours per Day Plants.

50	High-speed tandem compound, non-condensing, D. C.	69	2.062	1.449	3.327	3.749	4.112	55	3.908	1.818	4.818	5.272	5.727
75	...do.	71	2.029	1.409	2.738	3.085	3.438	57	2.962	1.754	3.839	4.278	4.717
100	...do.	77	1.681	1.299	2.331	2.665	2.980	62	2.447	1.613	3.263	3.656	4.059
100	High-speed tandem compound, condensing, D. C.	83	1.542	1.320	2.152	2.457	2.762	66	2.293	1.515	3.061	3.430	3.808
150	..do.	82	1.164	1.220	1.774	2.079	2.384	66	1.728	1.515	2.485	2.864	3.243
300	Cross compound, condensing, direct connected	123	.835	.813	1.242	1.445	1.648	99	1.238	1.010	1.743	1.995	2.248
500	.do.	130	.732	.769	1.117	1.309	1.501	105	1.087	.952	1.563	1.801	2.039
750	..do.	137	.699	.730	.964	1.146	1.328	110	.887	.909	1.341	1.569	1.796
1,000	.do.	144	.567	.694	.915	1.088	1.263	116	.841	.862	1.272	1.488	1.703
1,500	..do.	153	.529	.664	.856	1.019	1.183	123	.786	.813	1.191	1.394	1.596
2,500	..do.	162	.550	.617	.859	1.013	1.167	151	.810	.662	1.141	1.307	1.472
5,000	..do.	170	.470	.588	.773	.920	1.067	162	.704	.617	1.013	1.167	1.322
10,000	.do.	175	.438	.571	.723	.866	1.009	165	.644	.606	.947	1.099	1.250
20,000	Manhattan type, 4-cylinder, condensing, D. C.	179	.389	.559	.668	.808	.948	168	.571	.595	.869	1.018	1.166

10 Hours per Day Plants.

50	High-speed tandem compound, non-condensing, D. C.	63	4.095	1.587	4.830	5.227	5.623	50	5.909	2.000	6.969	7.469	7.969
75	...do.	65	3.111	1.539	3.881	4.265	4.650	51	4.586	1.961	5.566	6.066	6.546
100	do.	71	2.602	1.109	3.306	3.658	4.010	56	3.827	1.780	4.720	5.167	5.613
100	High-speed tandem compound, condensing, D. C.	74	2.520	1.351	3.196	3.533	3.871	59	3.769	1.695	4.607	5.031	5.454
150	do.	74	1.934	1.251	2.610	2.947	3.285	59	2.882	1.695	3.730	4.154	4.577
300	Cross compound, condensing, direct connected.	112	1.979	.893	2.426	2.649	2.872	88	2.954	1.186	3.522	3.806	4.090
500	.do.	119	1.478	.840	1.898	2.108	2.318	93	2.206	1.075	2.744	3.013	3.282
750	...do.	124	1.165	.807	1.568	1.790	1.991	98	1.767	1.020	2.277	2.532	2.787
1,000	do.	131	1.025	.763	1.406	1.597	1.788	104	1.527	.962	2.008	2.248	2.489
1,500	..do.	139	.973	.719	1.332	1.512	1.692	110	1.450	.909	1.904	2.131	2.359
2,500	. do.	147	.952	.690	1.292	1.462	1.632	133	1.413	.756	1.791	1.981	2.170

to be 10 per cent of total in all condensing plants. Cost of water and

by the average daily load curve, and must not be confounded with
lay plants only.
plant without overload.
umption in any of the above plants, particularly with the lower load

factors

TABLE 105.

COST OF POWER.

Examples of Isolated Station Practice.

	Large Office Building.	Small Office Building.	Manufacturing Plant, Electroplating.
Rated capacity...............kilowatts	500	50	500
Yearly capacity.........kilowatt hours	4,380,000	438,000	4,380,000
Actual load.............kilowatt hours	670,000	40,820	3,500,000
Yearly load factor.............per cent	17	9.1	80
Curve load factor..............per cent	17	24.7	80

Operating Charges, per Year.

Labor..................................	$6,050.00	$1,400.00	$12,300.00
Coal and ashes.........................	6,342.00	960.00	9,100.00
Water..................................	642.00	75.00	....
Oil and waste..........................	168.00	90.00	210.00
Lamps..................................	395.00	41.00	50.00
Repairs and renewals...................	69.00	182.00	1,008.00
Total............................	$13,666.00	$2,748.00	$22,668.00

Fixed Charges, per Year.

Interest (5 per cent)..................	$3,500.00	$325.00	$4,500.00
Depreciation (6 per cent)..............	4,200.00	628.00	5,400.00
Insurance (½ per cent).................	350.00	30.00	450.00
Taxes (1½ per cent)....................	1,050.00	90.00	1,350.00
Rental.................................	900.00	....	....
Total............................	$10,000.00	$1,073.00	$11,700.00

Cost per Kilowatt Hour, Cents.

Operating charges......................	1.79	6.78	.65
Fixed charges..........................	1.32	2.62	.33
Total cost.............................	3.11	9.40	.98

TABLE 106.

COST OF POWER — ISOLATED STATIONS.

(I. D. Parsons, *Engineering Magazine*, January–February, 1902.)

Building number	1	2	3	4	5	6	7	8
Type of building	Office	Office	Office	Office	Office	Loft	Loft	Loft
Number of floors	11	Over 20	13	15	12	12	12	10
Ground plan	50×120	70×100	60×150	75×100	40×100	200×176	100×100	40×200
Number of electric elevators	4–25 H.P.	10–25 H.P.	6–27 H.P.	3–20 H.P.	8–20 H.P.	15–20 H.P.	4–20 H.P.	4–25 H.P.
Part of building lighted	All	All	All	All	6 floors	All	All	All
Type of engine	Simple	Compound	Compound	Simple	Simple	Simple	Simple	Simple
Number of generators	4	4	5	3	1	3	3	3
Total capacity of generators, kilowatts	260	575	475	300	40	300	260	90
Average load on generators	.55	.75	.75	.90	.65	.90	.75	.60
Voltage	220 & 240	120	115 & 240	116	240	240	240	240
Type of boiler	H. Tub.	W. Tub.	W. Tub.	H. Tub.	H. Tub.	H. Tub.	H. Tub.	H. Tub.
Boiler pressure	90	125	115	100	95	90	100	100
Back pressure	0	0	1 to 3	4	1 to 5	3 to 9	2 to 5	7 to 10
Kind of coal	Pea	Buck.	Pea	Buck.	Buck.	Rice	Buck.	Buck.
Capacity storage battery, ampere hours	....	3,500	....	3,000	300	....	....	....
Load factor of lighting load	.24	.43	.72	....	.87	.70	.45	....

Cost of Light, Heat, and Power.

Time covered by figures	1 year	1 year	9 months	1 year	1 year	1 year	1 year	1 year	1 year
Labor	$2,460	$7,228	$6,750	$9,000	$6,318	$1,484	$4,628	$4,776	$1,662
Coal and removal of ash	2,758	10,070	7,663	10,238	9,174	1,012	4,192	2,736	2,129
Water	254	776	756	1,004	906	97	446	204	270
Oil and waste	181	287	171	215	128	103	193	108	80
Lamps	120	600	338	450	420	44	233	33	57
Repairs and sundries	183	50	429	568	65	295	100	108	478
Interest and depreciation	1,340	10,420	3,630	4,840	5,305	1,274	2,200	2,730	908
Central station service	1,370	....	....	....	....	225	....	300	579
Total	$8,666	$29,431	$19,737	$26,315	$22,370	$4,534	$11,992	$10,995	$6,163
Output — light, kilowatt hours	91,804	409,233	395,949	550,000	....	18,123	....	....	....
Output — power, kilowatt hours	59,760	266,344	160,963	225,000	....	48,437	....	....	....
Output — total kilowatt hours	151,564	677,577	556,912	775,000	721,30	66,560	379,939	143,320	130,000
Cost per kilowatt hour	$0.057	$0.044	$0.034	$0.034	$0.031	$0.068	$0.032	$0.077	$0.047
Tons of coal consumed	919	3,007	2,250	3,012	2,019	337	1,873	863	835
Pounds coal per kilowatt hour	15.5	10.0	9.1	....	8.2	11.4	11	13.5	14.4

TABLE 107.

COST OF POWER — ISOLATED STATIONS.

(I. D. Parsons, *Engineering Magazine*, January–February, 1902.)

Building number	10	11	12	13	14	15		16	17
Type of building	Ap't H'se	Ap't H'se	Ap't H'se	Hotel	Hotel	Club		Club	Club
Number of floors	7	10	11	6	9	5		3	4
Ground plan	90×65	44×81	50×200	100×200	25×60	35×100		75×200	110×100
Number of electric elevators	1–15 H.P.	2–20 H.P.	3–25 H.P.	....	2–15 H.P.	....		1–20 H.P.	....
Floors of building lighted	All	All	All	1st & 2d	All	All		All	All
Number of generators	2	2	3	2	2	1		3	2
Total capacity of generators, kilowatts	45	69	175	66	80	20		130	50
Average load on generators	.48	.20	.60	.50	.67	.40		.40	.75
Voltage	115	240	115	118	115	116		118	118
Type of boiler	H. Tub.	Marine	W. Tube	H. Tub.	H. Tub.	H. Tub.		H. Tub.	H. Tub.
Boiler pressure	90	100	125	90	100	80		90	80
Back pressure	5	2 to 5	3 to 5	4	3 to 6	6		3 to 5	3 to 5
Kind of coal	Pea	Buck.	Buck.	Pea	Bitum.	Buck.		Buck.	Buck.
Load factor of lighting load	.43	.40	.40	.60	.67	.50		.44	.85
Cost of Light, Heat, and Power.									
Time covered by figures	1 year	6 mos.	1 year	10 mos.	1 year	5 mos. '00	5 mos. '01	1 year	1 year
Labor	$2,280	$1,320	$3,840	$480	$3,690	$325	$325	$2,700	$2,400
Coal and removal of ashes	1,995	936	5,563	620	5,530	263	389	1,355	1,551
Water	164	64	303	60	486	22	30	117	196
Oil and waste	136	72	240	92	339	6	6	56	90
Lamps	28	8	136	40	354	5	6	24	93
Repairs and sundries	45	46	200	50	500	16	16	112	90
Interest and depreciation	572	495	2,376	880	1,045	115	115	1,100	770
Central station service	....	300	....	....	....	....	....	....	584
Total	$5,220	$3,241	$12,658	$2,222	$11,944	$752	$887	$5,464	$5,774
Output — light, kilowatt hours	55,730	....	182,360	115,000	....	6,306	9,407	....	68,513
Output — power, kilowatt hours	26,400	....	50,000	....	....	....	....	....	....
Output — total kilowatt hours	82,130	26,605	232,960	232,960	115,000	6,306	9,307	93,800	68,513
Cost per kilowatt hour	$0.064	$0.120	$0.055	$0.019	$0.037	$0.12	$0.09	$0.057	$0.084
Tons of coal consumed	570	312	1,919	1,007	1,826	75	111	466	486
Pounds coal per kilowatt hour	15.5	25.1	18.4	13.5	12.4	26.7	26.4	11.1	16.6

TABLE 107a.

YEARLY OPERATING COSTS IN FOUR TYPICAL CENTRAL STATIONS, STATE OF MASSACHUSETTS

Year ending June, 1909.

	Salem Electric Light Co.	Fitchburg Gas & Electric Co.	Haverhill Electric Co.	Malden Electric Co.
Type of Prime Mover	6 Engines	3 Engines	2 Turbines, 1 Engine	1 Turbine, 3 Engines
Rated station capacity, kw.	2500	2000	2300	....
Output, millions of kw. hrs.	3.106	4.006	3.721	4.715
Yearly load factor, per cent.	14.2	22.9	18.5	....
Total station operating force	14	12	13	14
Cost of fuel, dollars per ton	4.51	4.52	3.97	3.78
Coal per kw. hr.	3.3	3.28	3.27	3.02
Operating Costs, Cents per Kilowatt Hour.				
Coal	0.740	0.740	0.650	0.565
Oil and waste	0.025	0.015	0.190	0.020
Water	0.027	0.025	0.003	0.045
Wages	0.410	0.308	0.285	0.320
Station building repairs	0.034	0.017	0.063	0.023
Steam equipment repairs	0.158	0.041	0.073	0.072
Electrical equipment repairs	0.011	0.072	0.019	0.14
Miscellaneous	....	0.024	0.040	0.21
Total	1.412	1.242	1.152	1.08

TABLE 107b.

COST OF POWER, CENTS PER KW. HOUR. STEAM-ELECTRIC CENTRAL STATIONS.

Year ending June 30, 1908.

	Boston.	Worcester.	Lowell.	Fall River.	Malden.	Cambridge.	Lynn.
Fuel	.462	.703	.710	.880	.635	.690	.618
Oil and waste	.008	.027	.009	.032	.017	.019	.012
Water	.024	.034	.008	.012	.032	.055	.040
Wages	.192	.360	.262	.538	.342	.347	.296
Station repairs	.015	.012	.020	.012	.035	.021	.052
Steam repairs	.042	.055	.020	.037	.072	.059	.147
Electrical repairs	.056	.055	.009	.029	.014	.046	.045
Miscellaneous	.023	.000	.022	.080	.033	.000	.000
Total	.822	1.246	1.060	1.620	1.180	1.237	1.210
Cost of fuel per ton	3.99	4.79	4.75	4.68	4.49	4.40	3.60
Output, millions kilowatt hours per year	88.5	5.4	9.4	4.0	4.6	6.0	8.7
Capacity of station, thousands of H.P.	73.5	5.90	7.39	4.43	4.87	6.75	8.2

TABLE 108.

COST OF POWER (1907), FIRST NATIONAL BANK BUILDING, CHICAGO.

Total weight of coal burned .... tons	14,956
Total weight of water evaporated .... gallons	22,100,000
Total electrical output .... kilowatt hours	1,546,600
Water actually evaporated per pound of coal as fired	6.1

Electric-Light Plant. (Cost of Power only.)

Pounds of coal per kilowatt hour	6.57
Cost of coal per kilowatt hour .... cents	0.88
Cost of labor per kilowatt hour .... cents	0.78
Cost of supplies per kilowatt hour .... cents	0.11
Total	1.77

All expenses of entire plant charged against switchboard.

Pounds of coal per kilowatt hour	19.34
Cost of coal per kilowatt hour .... cents	2.23
Cost of labor per kilowatt hour .... cents	1.59
Cost of supplies per kilowatt hour .... cents	1.18
Total	5.00

Elevator Plant.

Passengers handled	2,016,300
Total car miles	92,700
Cost of labor per car mile .... cents	2.02
Cost of material per car mile .... cents	3.12
Cost of power* per car mile .... cents	3.00
Total	8.14

* Approximate.

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Cost of Steam and Electricity, *Engineering*	74: 667	Nov. 21, 1902
Comparative Cost of Steam and Water Power (W. O. Webber), *Engineering Magazine*	15: 923	Sept., 1898
Steam Costs in an Industrial Combination (W. D. Ennis), *Engineering Magazine*	28: 86	Oct., 1904
Fuel Economy of Engines in Electric Railway Power Stations (Carpenter), *Engineering News*	42: 234	Oct. 12, 1899
Reduction in Cost of Steam Power from 1870 to 1897, *Engineering Record*	37: 12	Dec. 4, 1897
Economy in Use of Coal for Production of Power (I. N. Hollis), *Engineering Record*	46: 491	Nov. 22, 1902
Economy of Fuel in Electric Plants (Edit), *Engineering Record*	48: 233	Aug. 29, 1903
Economical Steam Making (Edit), *Engineering Record*	48: 385	Oct. 3, 1903
Cost of Fuel and Power in the South, *Power*	18: 13	Nov., 1898
Economical Production of Steam with Special Reference to the Use of Cheap Fuel, *Power*	19: 19	June, 1899
Suggestions for Steam Economy (W. M. Kay), *Engineer* (United States)	42: 655	Oct. 2, 1905
Yearly Cost of One Steam Horse Power, *Machinery*	9: 374	March, 1903

WATER.

Cost of Water and Electric Power (G. E. Walsh), *American Electrician*	16: 331	July, 1904
Comparative Cost of Power Generated by Steam Engine, Water Turbine, and Gas Engine, *Engineer* (London)	88: 322	Sept. 29, 1898
Investigation of the Cost of Power, *Engineer* (United States)	39: 64	Jan. 15, 1902
Comparative Cost of Generating Power by Steam Engine, Water Turbine, and Gas Engine (J. B. C. Kershaw), *Engineering*	70: 351	Sept. 14, 1900
	390	Sept. 21, 1900
Cost of Water Power in France, *Engineering*	76: 571	Oct. 23, 1903
Comparative Cost of Steam and Water Power (W. O. Webber), *Engineering Magazine*	15: 923	Sept., 1898
Cost of Hydraulic Transmission of Power (E. B. Ellington), *Engineering Magazine*	17: 233	May, 1899
Cost of Hydraulic Power in Switzerland, *Engineering Record*	41: 182	Feb. 24, 1900
Analysis of the Commerical Value of Water Power (A. F. Nagle), *Engineering Record*	46: 540	Dec. 6, 1902
Costs of Pumping Water, *Power*	20: 12	Nov., 1900

COST OF POWER.
(1908–1910.)

Approximate Cost of Gas Power (M. P. Cleghorn), *Power and Engineer*		April 7, 1908
Central Station vs. Private Plants, *Engineering*		Feb. 26, 1909
Comparative Cost of Power Production, *Electrical Age*	40: 63	March, 1909
Electrical World	53: 792	April 1, 1909
Electrical Review and Western Electrician	55: 773	April 28, 1909
Cost of a Gas Engine and of a Combined Steam Plant, *Engineering Record*	60: 272	Sept. 4, 1909
Cost of a Kilowatt-Hour (R. A. Day), *Electrical World*	54: 853	Oct. 7, 1909
Cost of Power (H. G. Stott), Pro. Am. Inst. Elec. Engrs.	28: 283	April, 1909
Cost of Power in a 3000 Kw. Turbine Plant, *Electrical Review and Western Electrician*	55: 62	Oct. 2, 1909
Cost of Power in a 1500 Kw. Central Station, *Engineering News*	61: 471	April 29, 1909
Cost of Power in Small Plants (W. E. Snow), *Engineering Magazine*	33: 169	May, 1908
Cost of Power in Four Central Stations, *Electrical World*	55: 813	March 31, 1910
Cost of Power for Various Industries (C. T. Main), Jour. Assoc. Eng. Soc.	44: 151	March, 1910
Engineering Record	60: 711	Dec. 25, 1909
Cost of Power in Varying Units (W. O. Webber), *Engineering Magazine*	35: 562	July, 1908
Cost Systems and Time Keeping, Columbus, O., Municipal Electric Lighting Plant, *Engineering News*		Dec. 3, 1908
Electric Power Costs in Small Stations, *Engineering Record*	59: 30	Jan. 9, 1909
First Cost of Plant and Cost of Generating and Distributing Electricity for Lights, Brooklyn Edison Co., *Engineering Contractor*	33: 393	April 6, 1910
Isolated Power Plant Costs and Their Relation to Central Station Service (W. F. Lyod), *Electrical World*	53: 323	Feb. 4, 1909
Isolated Station Records and Accounting Power		April 28, 1908
Operating Costs of Large Units, *Power and Engineer*	31: 981	May 31, 1910
Power Costs for Factories, *Engineering Record*	60: 604	Nov. 27, 1909
Power Plant Waste (P. R. Moses), *Cassier's Magazine*	36: 497 37: 12	Oct., 1909 Nov., 1909
Relation of Load Factor to Power Costs, Jour. Wes. Soc. Engr.	14: 241	April, 1909
Engineering Record	59: 702	June 5, 1909
Representative Data from Electric Power Plant Operation (H. S. Knowlton), *Engineering Magazine*	36: 833	Feb., 1909
Systems of Charging for Electrical Energy (W. T. Ryan), *Engineering Magazine*		April, 1909
The Small Station and its Economical Operation, *Western Electrician*	43: 10	July 4, 1908
The Valuation of Steam Power Plants (C. T. Main), *Electrical Age*	39: 228	Oct., 1908
Useful Figures from Practical Power Plant Operation, *Electrical World*	54: 781	Sept. 30, 1909
Working Results from a Gas-Electric Power Plant (J. R. Bibbins), Pro. Am. Inst. Elec. Engrs.	27: 1123	July 1, 1909

CHAPTER XVIII.

TESTING AND MEASURING APPARATUS.

403. General. — The importance of maintaining a system of records has been referred to in paragraph 388. The various items which may be recorded and the instruments and appliances used in this connection are outlined in the accompanying chart. In large stations a full complement of indicating, recording, and integrating instruments may prove to be a good investment if intelligently and closely studied by the operating engineer with a view to locating and eliminating unnecessary losses. The instruments should be inspected and calibrated at intervals, since many of them are delicately constructed and are apt to become inaccurate after a few months' service. Steam gauges, thermometers, and pyrometers, and particularly water meters are subject to appreciable error after considerable use. Voltmeters, ammeters, and other switchboard instruments are easily deranged, especially when subjected to continuous vibration or to high temperature.

404. Weighing the Fuel. — In most small plants the delivery tickets of the coal dealer are depended upon for the weight of coal used, no attempt being made to determine its evaporative value, and the economy of the plant is judged by the size of the coal bill. In such cases a considerable saving can be effected by keeping a daily record covering at least the coal and water consumption. The coal can be conveniently weighed on ordinary platform scales. In a number of large stations the weight of coal is determined by suspended weighing hoppers, which may be stationary, as in Fig. 109, or mounted on a traveling truck, as in Fig. 110. The scales of such devices are made indicating, autographic, integrating, or a combination of the three, the latter costing but little more than the simple indicating or recording devices.

405. Measurement of Water. — The most accurate means of measuring water is by the use of two or more tanks resting upon scales, arranged to be filled and emptied alternately. This method, however, involves considerably more time than is ordinarily at the disposal of the fireman or engineer. The usual practice is to place a hot-water meter on the pressure side of the feed pump, with provision for calibration without shutting off the feed supply to the boilers.

TESTING AND MEASURING APPARATUS

STEAM PLANT.

- **Weights**
  - Fuel
    - Platform scales, indicating and autographic.
    - Suspension hoppers, indicating and autographic.
  - Water
    - Platform scales and tanks.
    - Water meters
      - Piston, Rotary, Disk — Integrating.
      - Venturi, indicating and autographic.
    - Weirs.
  - Steam
    - Weighing condensed steam.
    - Steam meters
      - St. John's, autographic.
      - Burnham, indicating.
- **Pressures**
  - High
    - Bourdon gauge, indicating and autographic.
    - Manometers, mercurial, indicating.
  - Low
    - Manometers — mercurial, indicating, and autographic.
    - Manometers — water, indicating, and autographic.
    - Diaphragms, indicating and autographic.
- **Temperatures**
  - Up to 800 deg. F.
    - Mercurial thermometers, indicating.
    - Expansion thermometers, indicating and autographic.
  - 800 to 2500 deg. F.
    - Expansion thermometers, indicating and autographic.
    - Resistance thermometers, indicating and autographic.
    - Thermo-electric thermometers, indicating and autographic.
  - Over 2500 deg. F.
    - Optical pyrometer, indicating and autographic.
    - Platinum or clay ball pyrometer.
- **Power**
  - Indicated
    - Indicators, hand manipulated.
    - Indicators, continuous autographic.
  - Developed
    - Rope brake.
    - Prony brake.
    - Absorption dynamometers.
    - Electric generator.
- **Flue gas analysis**
  - Orsat apparatus.
  - Arndt's econometer, indicating.
  - Sarco and Ados recorder, autographic.
  - Uehling gas composimeter, autographic.
- **Moisture**
  - In air............Hygrometer, indicating and autographic.
  - In steam.........Calorimeters....
    - Separating.
    - Throttling.
- **Fuel analysis**
  - Coal calorimeters.
    - Mahler bomb.
    - Carpenter.
    - Thompson.
    - Parr.
  - Gas calorimeter.....Junker.

ELECTRICAL PLANT.

Voltage.......Voltmeters, A.C. and D.C., indicating and autographic.
Current.......Ammeters, A.C. and D.C., indicating and autographic.
Output........Wattmeters, A.C. and D.C., integrating and autographic.
Power factor ..Power factor meters, A.C. only, indicating and autographic.
Frequency.....Frequency meter, A.C. only, indicating.
Synchronism ..Synchronizers, A.C. only, indicating.

There are several types of meters in common use. Fig. 419 illustrates the *piston* type, in which reciprocating pistons are displaced by a definite volume of water; Fig. 305, the *rotary* type, depending upon

Fig. 419. A Typical Piston Water Meter.

the displacement of rotating impellers; Fig. 420, the *disk* type, in which impellers are given a combined rotating and tilting motion. When periodically calibrated, water meters give satisfactory results. When graduated to read in pounds the accuracy depends upon the temperature range of the water; thus the density of water at 62 degrees F. is 62.36 pounds per cubic foot and at 212 degrees it is 59.76, a range of 2.6 pounds per cubic foot. Hence a meter calibrated to read correctly at 62 degrees F. will have an error of about 4.2 per cent if used to measure water at 212 degrees. The average range in feed temperature is seldom greater than 40 degrees, and if the meter is calibrated for the mean temperature the error is somewhat less than one per cent.

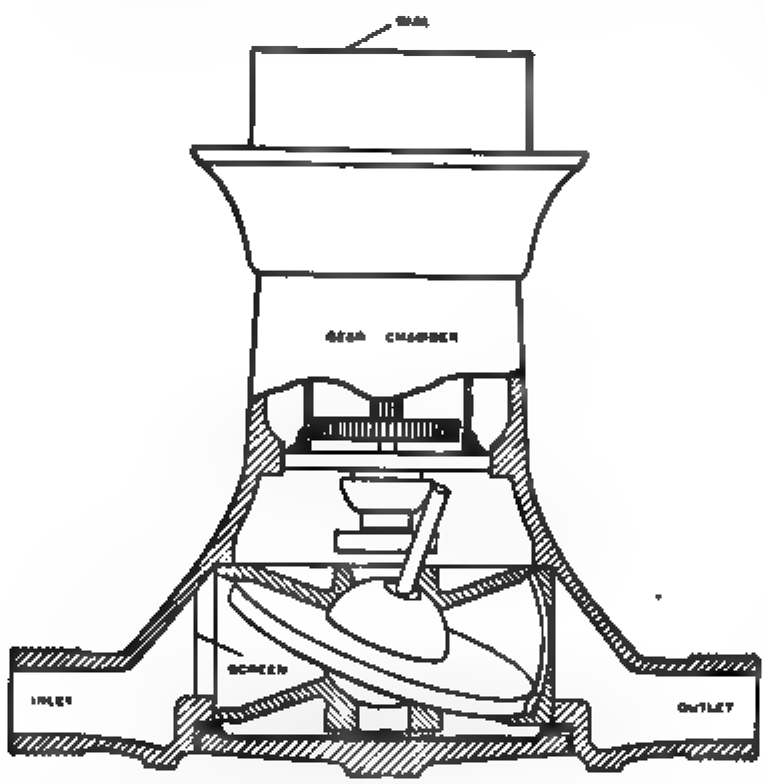

Fig. 420. A Typical Disk Water Meter.

The *Venturi* meter, Fig. 421, is frequently employed for measuring large volumes of water, as in city waterworks, and in connection with condensing plants.* It amounts practically to a constriction in the diameter of the pipe, is readily installed, and the total absence of

* Tests on a Venturi Meter for Boiler Feed. Proc. A.S.M.E., Mid.-Oct., 1909, p. 1065.

working parts is a great advantage. The meter is supplied with either indicating or autographic manometer. With water at constant temperature the error in the readings should not exceed one per cent.

The *pitometer* is a simple adaptation of the well-known pitot tube, and is used for measuring the flow of water through pipes where it is

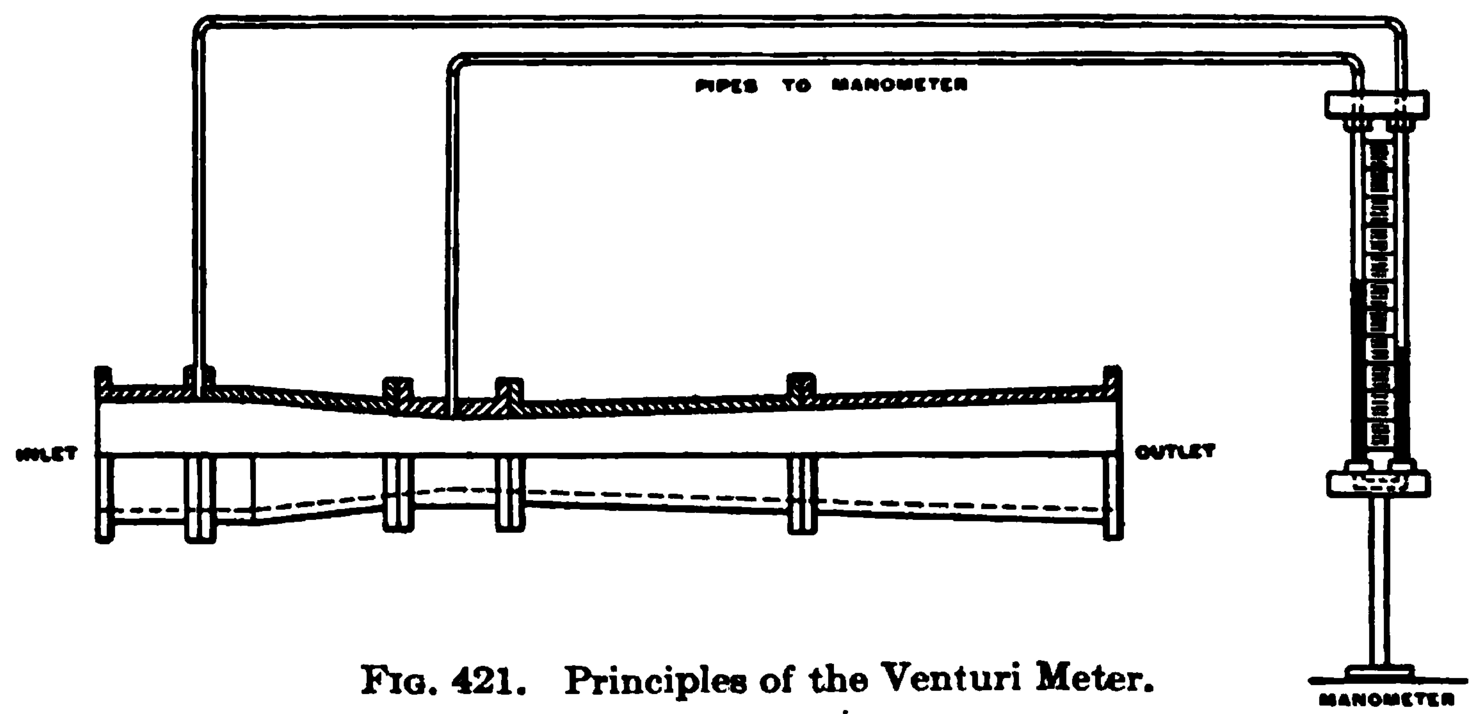

FIG. 421. Principles of the Venturi Meter.

impracticable to insert a meter. It is only necessary to drill a small hole in the pipe for the introduction of the tube. The volume flowing may be calculated from the readings of the manometer or may be autographically recorded.

In measuring large volumes of water flowing in open channels the measurements are made by weirs of suitable proportions or by current meters.

Water Measurement, General: Eng. Rec., Feb. 15, 1902.

Water Meters: Trans. A.S.M.E., 18–134, 14–676, 5–63; Engng. News, Jan. 3, 1907, March 9, 1905, June 16, 1904, p. 569; Eng. Rec., Nov., 1903; Stevens Ind., Jan., 1901; Jour. New Eng. Waterworks Assn., June, 1907; Elecn., Lond. May 8, 1908.

Venturi Meter: Revue Technique, Feb. 10, 1905, Eng. News, Feb. 28, 1901; Prac. Engr., Feb. 15, 1907; Pro. A.S.M.E., Nov., 1906; Trans. A.S.C.E., Nov., 1907, 57–531 (1906); Engng., Feb. 22, 1907, p. 236.

Pitometer: Jour. Franklin Inst., Dec., 1907, p. 425; Tech. Quar., June, 1907; Jour. New Eng. Waterworks Assn., June, 1906; Trans. A.S.M.E., 25–184; Sib. Jour. Engng., May, 1902; Jour. Assn. Eng. Soc., Aug., 1901; Eng. News, March 31, 1904, Dec. 21, 1905; Pro. Engrs. Soc. of West. Penn., Dec., 1906.

Pitot Tube: Am. Mach., Aug. 9, 1906, p. 175; Trans. A.S.C.E., 47–6, 57–265, 25–184; Eng. News, March 31, 1904, p. 318, Dec. 21, 1905, p. 660; Progressive Age, June, 1906, p. 63; Jour. Assn. Eng. Soc., Aug., 1901, p. 35; Sib. Jour., May, 1902; Cal. Jour. of Tech., May, 1905.

Weirs: Engr. (Lond.), June 5, 1903, p. 562, Aug. 17, 1906; Am. Soc. Civ. Engrs., 44–160; Eng. Rec., July 13, 1901, p. 32.

Current Meters: Cal. Jour. of Tech., April, 1904; Eng. News, March 7, 1907, p. 263, Feb. 12, 1902; Eng. Rec., Dec. 19, 1903; Pro. Am. Soc. Civ. Engrs., Sept., 1901, Nov., 1901, Dec., 1901.

Piezometers: Power, Aug., 1907, p. 569; Pro. Am. Soc. Civ. Engrs., 44–34, 49–112, 51–252; Eng. News, Sept. 13, 1906, p. 271; Power, Aug., 1907, p. 569.

406. Measurement of Steam. — In surface-condensing plants the weight of steam consumed by the engines is conveniently obtained by weighing the condensed steam; in practice, however, this method

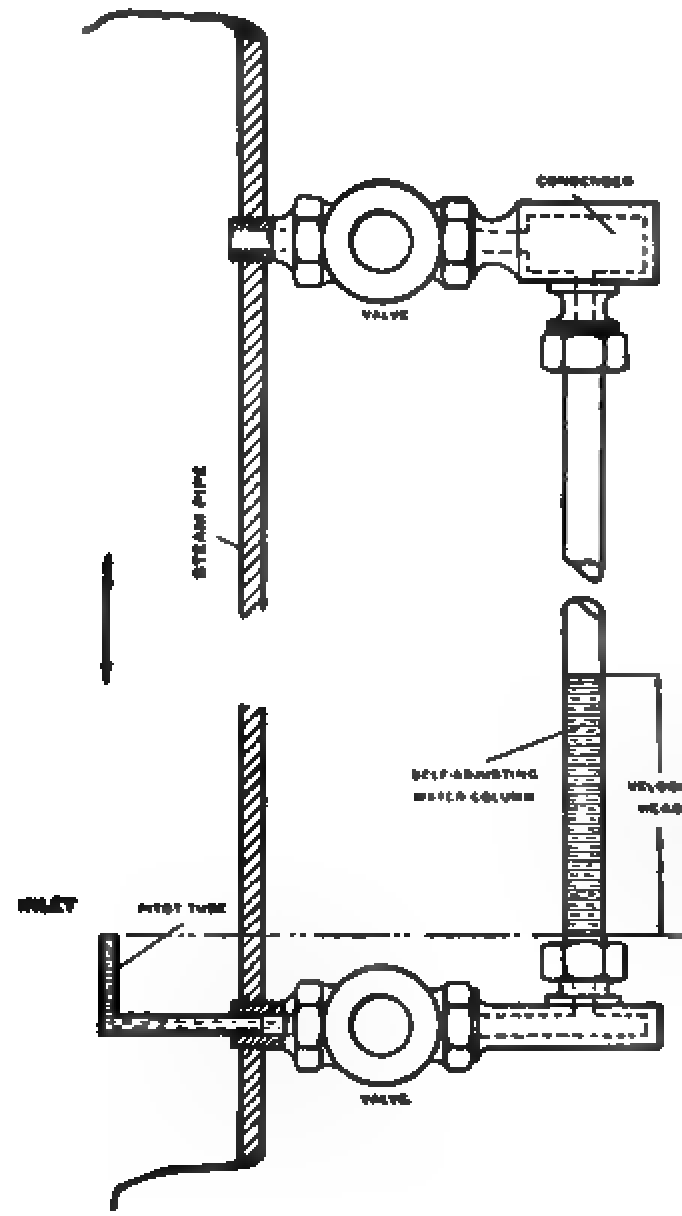

FIG. 422. St. John's Steam Meter.

FIG. 422a. Principles of the Burnham Steam Meter.

is adopted only when testing the plant, the feed-water measurement sufficing for general recording purposes.

When steam is supplied to various points and the weights cannot be readily determined by condensing, steam meters are sometimes used. The St. John's, Fig. 422, is the best known in this country. This apparatus records the weight of steam passing through the seat of an

automatically lifting valve which rises and falls as the demand for steam increases or diminishes. When the maximum fluctuation in steam pressure is less than 10 pounds per square inch and the moisture in the steam is practically constant, this apparatus is said to register within two per cent of the actual weight flowing.

The Burnham steam meter, Fig. 422a, a recently patented device, offers the advantages of low first cost and simplicity of installation. This apparatus is attached to the steam pipe in a manner similar to a simple hydrostatic lubricator and occupies about the same space. It is based on the principle of the pitot tube, and the weight of steam flowing per unit of time is read from a graduated scale. Half-inch pipe fittings are used in connecting up.

In Europe the principles of the Venturi meter have been successfully applied to the measurement of steam. (Gluckauf, Dec. 9, 1905.)

Steam Meters: Proc. A.S.M.E., Mid.-Nov., 1909, p. 1239.

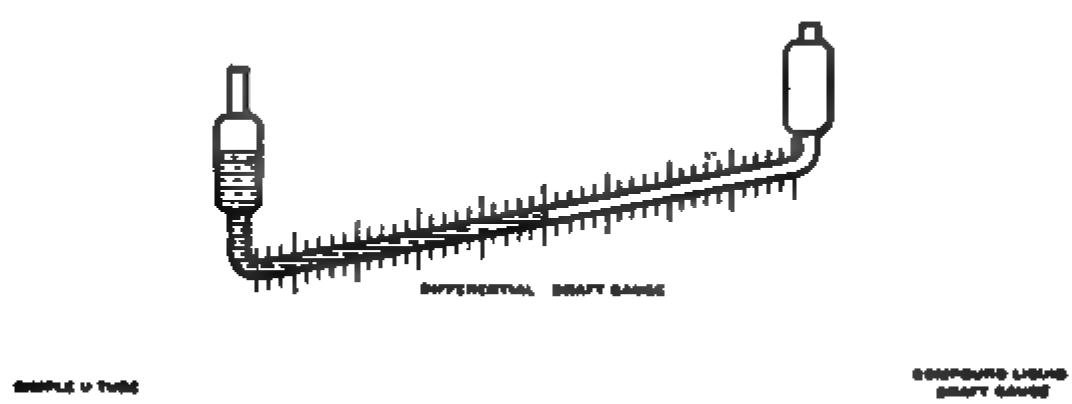

FIG. 423. Different Forms of Manometer Pressure Gauges.

406a. Pressure Gauges. — The Bourdon type of gauge, either autographic or indicating (Fig. 424), is the most familiar and satisfactory means of measuring pressures up to 1500 pounds per square inch or more, although diaphragm gauges are also used and both are employed as vacuum gauges. For the latter purpose, however, the mercurial vacuum gauge has the advantage of greater accuracy and is not subject to derangement. Bourdon gauges should be frequently standardized by comparison with a gauge of known accuracy, a mercury column, or a gauge tester.

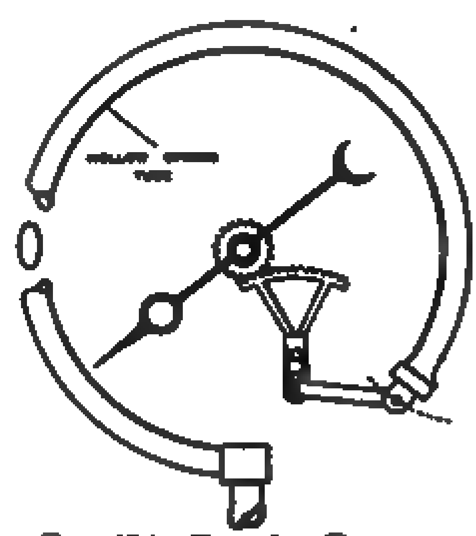

FIG. 424. Bourdon Pressure Gauge.

For measuring very low pressures, such as are found in boiler flues

or gas mains, indicating or recording diaphragm gauges may be had, but some form of U tube manometer is generally employed, the design best adapted to the purpose depending upon the accuracy required. The simple U tube (Fig. 423) when filled with mercury may be used for pressures limited only by the inconvenience due to length of tubes, or, with water as the fluid, for pressures only a fraction of an ounce per square inch. Where greater accuracy is required than can be obtained with the simple U tube, some modification may be employed, such as the Eames draft gauge with one inclined leg which magnifies the reading several times. A form of sensitive gauge is sometimes used which depends upon the use of two fluids of different specific gravity, as oil and water.

Pressure Gauges, General References: Mech. Engr., Aug. 17, 1907; Am. Elecn., July, 1901; Engng., Aug. 23, 1907; Elec. World, Feb. 2, 1907, p. 258; Power, March, 1905, p. 184.

Recording Pressure Gauge: Trans. A.S.M.E., 11–225, 14–325; Elec. World, April 28, 1906, p. 886.

Draft Gauges: Trans. A.S.M.E., 21–123; Engr. U.S., Feb. 15, 1907, p. 218; Mech. Engr., Oct. 27, 1906.

407. Measurement of Temperature. — For power-plant purposes mercurial thermometers are most convenient for measuring temperatures up to 400 degrees F., and are inexpensive. For higher temperature, up to say 800 degrees F., they are also adapted, but must be made of special glass and the space above the mercury filled with nitrogen under pressure to prevent vaporization of the mercury. Such thermometers must be used intelligently, and should be standardized from time to time, since they are subject to considerable change. The Bureau of Standards at Washington, D.C., is prepared to furnish certificates for which a nominal charge is made.

Fig. 425 shows a form of thermometer which is much used where a continuous autographic record is required. It depends for its operation upon the pressure produced by a fluid, liquid or gaseous, contained in a small bulb and exposed to the temperature to be measured. The pressure is transmitted to the recording mechanism through a flexible capillary tube which may be of considerable length. Such thermometers are suitable for feed water, flue gas, and temperatures not exceeding 1000 degrees F.

Fig. 426 illustrates a form of electrical pyrometer employing thermocouples which has come into wide use as a reliable means of measuring temperatures up to 2600 degrees F. The couples most frequently used are composed of platinum and platinum-rhodium, platinum and platinum-iridium, copper and copper-constantan, and copper and nickel,

the first named being adapted to the higher ranges of temperature. The electro-motive force set up, when the thermo-junction is heated, is proportional to the temperature and is measured by means of a sensi-

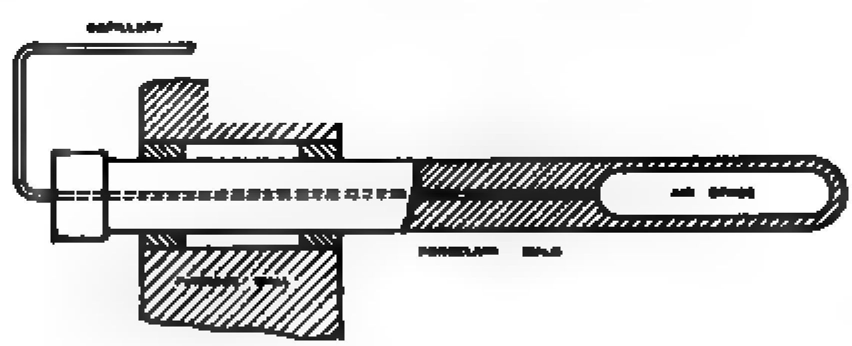

FIG. 425. Bristol Recording Pyrometer.

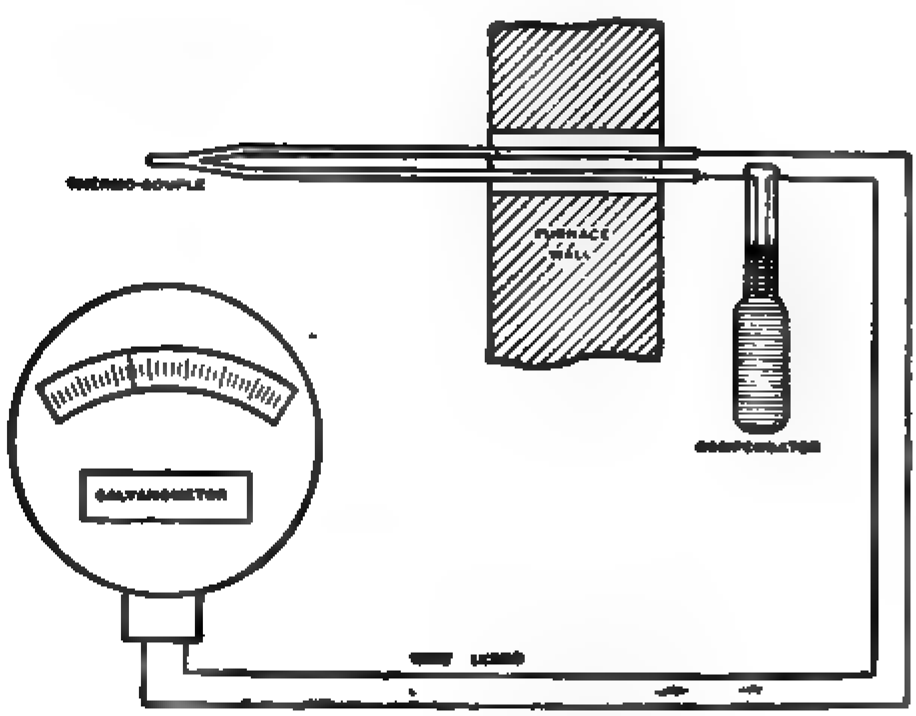

FIG. 426. Bristol Thermo-Electric Pyrometer.

tive millivoltmeter which is usually graduated to read temperature directly. Thermo-couples may be made to give an autographic record by means of a *thread recorder*.

Fig. 427 shows the element of an electrical thermometer based upon the change in resistance of a platinum wire when subjected to change in temperature. The resistance, in terms of temperature, is measured by a *Whipple indicator*, a convenient and portable form of Wheat-

FIG. 427. Element for Callendar Resistance Pyrometer.

stone bridge, or may be autographically recorded by means of a *Callendar recorder*. Resistance thermometers of this type are very sensitive and accurate, not easily deranged, and are limited in range only by the fusing points of the platinum and the porcelain protecting sheath.

For higher temperatures and for obtaining the temperatures of inclosed spaces above about 900 degrees F., such as boiler furnaces, annealing ovens, and kilns, various forms of *optical* and *radiation pyrometers* have been devised. In such devices no part of the instru-

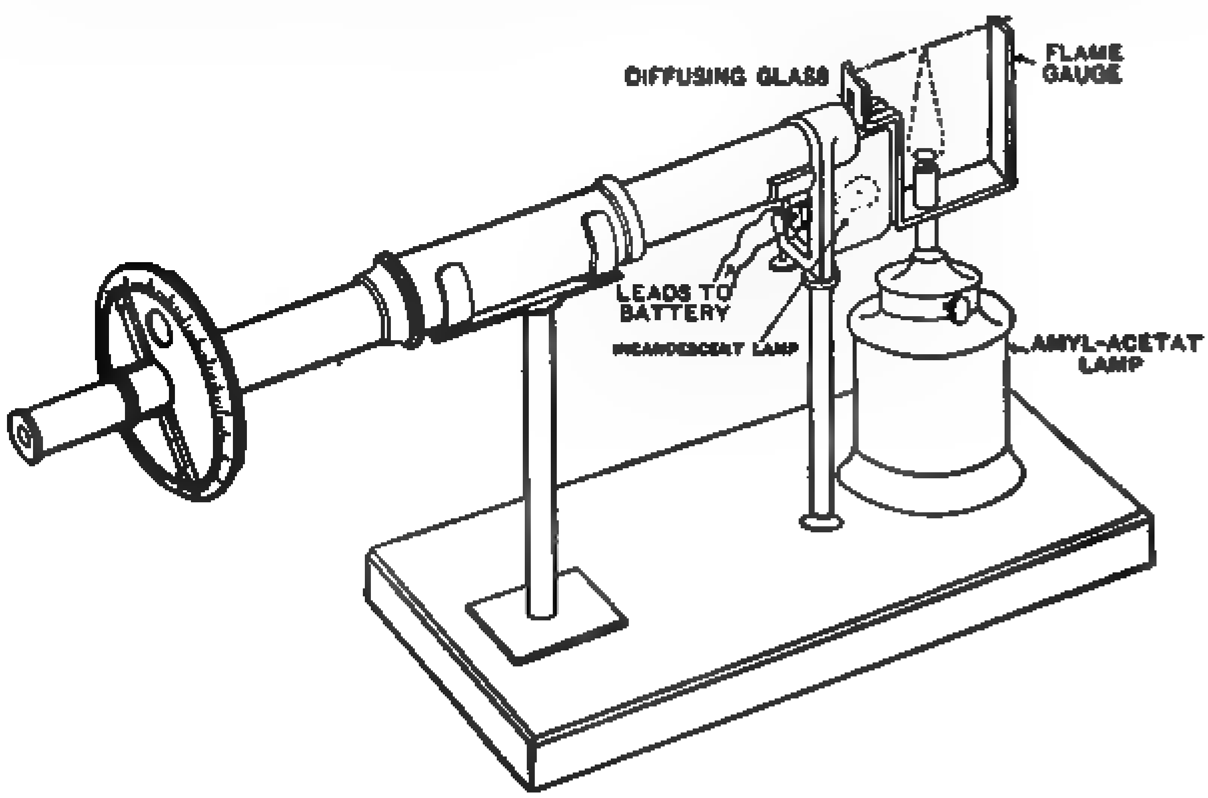

FIG. 428. Wanner Optical Pyrometer in Position for Standardizing.

ment is exposed to the temperature to be measured and hence suffers no injury from this cause. Optical pyrometers are based upon the measurement of the brightness of the hot body by comparison with a standard. The Wanner optical pyrometer is shown in Fig. 428.

After standardizing by comparison with an amyl-acetate lamp, it is only necessary to focus the instrument upon the source of heat to be measured and the temperature is read on the graduated scale.

TABLE 109.

TYPES OF THERMOMETERS IN GENERAL USE.

Principle of Operation.		Type.	Range in Degrees F. for which they can be used.
Expansion	Those depending on the change in volume or length of a body with temperature.	Gas	−400 to +2900
		Mercury, Jena glass, and nitrogen	−35 to +950
		Glass and petrol ether.	−325 to +100
		Unequal expansion of metal rods.	0 to 950
Transpiration and viscosity.	Those depending on the flow of gases through capillary tubes or small apertures.	The Uehling	0 to 2900
Thermo-electric	Those depending on the electro-motive force developed by the difference in temperature of two similar thermo-electric junctions opposed to one another.	Galvanometric	−400 to +2900
Electric resistance	Those utilizing the increase in electric resistance of a wire with temperature.	Direct reading on indicator or bridge and galvanometer.	−400 to +2200
Radiation	Those depending on the heat radiated by hot bodies.	Thermo-couple in focus of mirror.	300 to 4000
		Bolometer	−400 to Sun
Optical	Those utilizing the change in the brightness or in the wave length of the light emitted by an incandescent body.	Photometric comparison.	1100 to Sun
		Incandescent filament in telescope.	
		Nicol with quartz plate and analyzer.	
Calorimetric	Those depending on the specific heat of a body raised to a high temperature.	Platinum ball with water vessel.	32 to 3000
Fusion	Those depending on the unequal fusibility of various metals or earthenware blocks of varied composition.	Alloys of various fusibilities.	32 to 3350

Radiation pyrometers depend upon the measurement of the heat radiated from the hot body. The Fery radiation pyrometer, Fig. 429,

is the best-known instrument of this type. When focused upon the source of heat a cone of rays of definite angle is reflected by means of the mirror upon a thermo-couple located in its focus. The electromotive force set up is measured in terms of the temperature of the

FIG. 429. Féry Radiation Pyrometer.

source of heat by a millivoltmeter. Neither the couple nor any part of the instrument is ever subjected to a temperature much above 150 degrees F. The indications are practically independent of the distance from the source of heat, and the range is without limit.

Table 109 embodies in outline the principles and temperature ranges of the various types of thermometers in use. Temperature ranges verified by U. S. Bureau of Standards.

Indicating and Recording Thermometers, Expansion Type: Sci. Am. Sup., Dec. 16, 1905; Trans. A.S.M.E., 22–143; Jour. Am. Chem. Soc., 16–396; Jour. Soc. Chem. Ind., 13–61; Philosoph. Mag., 50–251, 1900.

Indicating and Recording Pyrometers, Thermo-Electric: Jour. West. Soc. Engrs., Sept., 1907; Cassier's Mag., Aug., 1905; Elec. Rec., Jan. 12, 1901; Elec. Chem. and Met., June, 1901.

Indicating and Recording Pyrometers, Electric-Resistance: Engng., May, 1899; Jour. Chem. Soc., 1890, 1895; Jour. Iron and Steel Inst., 1892; Pro. Royal Inst., Vol. XVI, 1901; Bureau of Standards, 3–641, 1907; Electrician, March 17, 1905, p. 880.

Indicating and Recording Pyrometers, Optical: Elecn., Lond., Aug. 17, 1906; Am. Mach., Vol. 28, 160–29; Sch. of Mines Quarterly, April, 1907; Bulletin No. 2, Bureau of Standards, Wash., 1905; Jour. de Phys., Sept., 1904; Engng., Sept. 6, 1907, Oct. 18, 1907; Cal. Jour. of Tech., Aug., 1907; Bureau of Standards. Bulletin No. 2; Iron Age, 73–24.

Miscellaneous References: Engng. Times, March, 1904; Engng., Feb. 17, 1903, March 6, 1904; Sci. Am. Sup., July 22, 1905; Min. Rept., Aug. 8, 1901; Iron Age, Feb. 7, 1907; Iron and Coal Tds. Rev., May 10, 1907; Am. Elecn., May, 1904; Physical Rev., 8–193; Roy. Soc. of Lond., 66–86, 1900.

408. Power Measurements. — The indicated horse power of reciprocating engines is usually obtained by means of the steam-engine indicator. There are several reliable types to be had, including the *continuous* indicator, which permits of several diagrams being taken successively on the same paper. Among other devices may be mentioned *mean pressure* indicators and those giving the horse power directly.

The developed horse power is determined by some form of *absorption dynamometer*. For description of such dynamometers see Appendix C, article XV, A.S.M.E. code for conducting steam-engine tests.

Power Measurements: Trans. A.S.M.E., 13-531; Am. Mach., Vol. 30, No. 27, Vol. 31, No. 5; Mechanical Engr., Feb. 23, 1907; Engng., June 14, 1907, p. 768.

Indicators, Continuous: Trans. A.S.M.E., 18-1020; Power, Jan., 1907, p. 26.

Prony Brakes: Trans. A.S.M.E., 15-62; Am. Mach., July 27, 1905, p. 127; Eng. News, Vol. 44, p. 216.

Water Absorption Dynamometers: Prac. Engr., Sept. 14, 1906; Trans. A.S.M.E., 11-958; Eng. News, Vol. 51, p. 475; Prac. Engr., Sept. 14, 1906, p. 326.

FIG. 430. Orsat Apparatus for Flue Gas Analysis.

409. Flue-Gas Analysis. — The simplest device for the analysis of flue gases is the *Orsat apparatus* (Fig. 430). In this apparatus a

measured volume, representing an average sample of the gas, is forced successively through pipettes containing solutions of caustic potash, pyrogallic acid, and cuprous chloride in hydrochloric acid, respectively, thus removing the carbon dioxide, the oxygen, and the carbon monoxide, the contraction of volume being measured in each case.

Orsat Apparatus: Trans. A.S.M.E., 18–901; Steam Boilers, Peabody and Miller, Chap. II; Power, Aug., 1907, p. 532; Engr. U.S., Jan. 1, 1907, p. 71.

For most practical purposes it is sufficient to determine the carbon dioxide. A simple and efficient device for continuously indicating the per cent of CO_2 is *Arndt's econometer*, Fig. 431. This apparatus

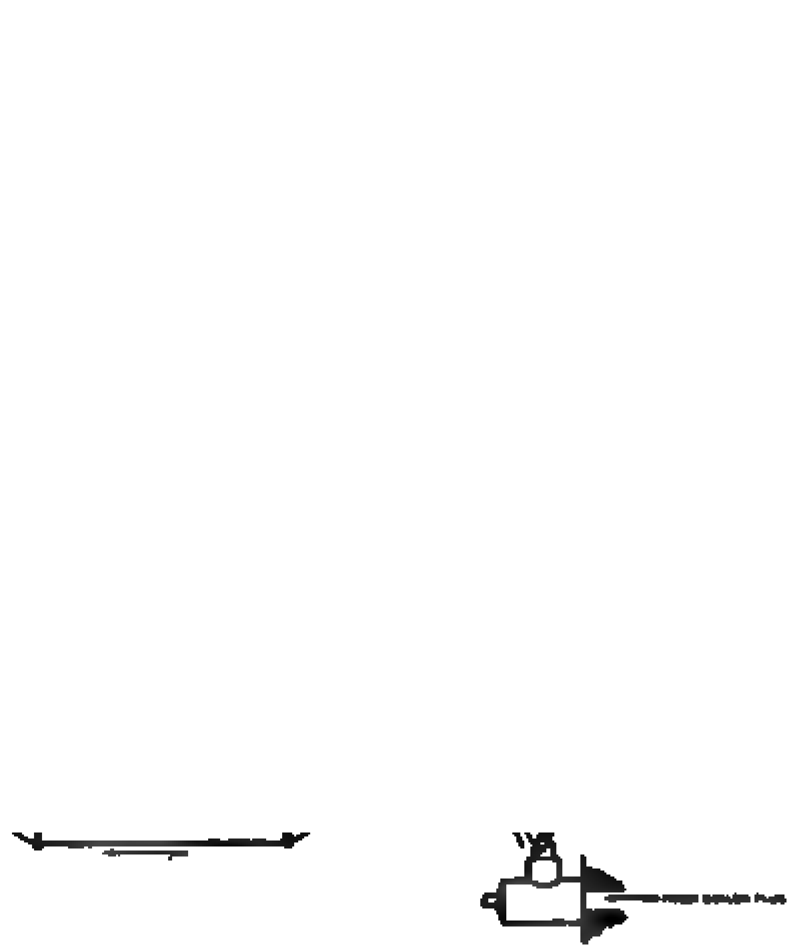

Fig. 431. Arndt's Econometer.

is a gas-weighing balance consisting essentially of a sensitive beam from one end of which is suspended a glass globe, closed at the top and open at the bottom, of about one pint capacity, and from the other end a compensating rod and scale pan. When not in operation the globe is filled with air and the scale pan and globe are in perfect balance, the indicator pointing to zero. When in operation the flue gases, thoroughly dried and filtered, are introduced in a continuous flow into the body of the hollow globe by means of a glass tube. The larger the per cent of CO_2 present in the contents of the globe the greater will be the deflection of the pointer, since CO_2 is about

50 per cent heavier than atmospheric air. The scale is graduated to read from 0 to 21 per cent CO_2, and the results obtained check closely with those of the Orsat apparatus. Arndt's econometer is not portable, though it may be placed almost anywhere where it can be easily seen by the fireman. When there are a number of boilers, and it is not desired to have a separate instrument for each, the econometer is connected with the breeching of each boiler by suitable piping, the gas from one boiler at a time being analyzed.

For descriptive details see circular issued by Joseph Wickes, 106 Fulton Street, New York.

FIG. 432. Ados CO_2 Recorder.

The *Ados recorder*, Fig. 432, *Sarco recorder*, Fig. 433, and *Uehling composimeter* are well-known types of instruments which give continuous autographic records of the percentage of CO_2 on clock-driven charts. These devices are very satisfactory, but are rather expensive and usually beyond the appropriation of small boiler plants.

The Sarco CO_2 recorder operates as follows: Gas is drawn continuously from the flue through a [illegible]-inch pipe by means of the aspi-

rator *Q*, and enters the apparatus through the tube *D*, flowing as indicated by the arrows through *C* and *E*. The aspirator requires from 6 to 8 gallons of water per hour, which is discharged into the pressure-regulating tank *L*. The greater portion overflows through *R*, and the balance is caused to issue in a fine stream by adjustment of cock *S* into *H* and *G* and thence to an upper compartment of vessel *K*, which it gradually fills, compressing the air which it contains and transmitting this pressure to the lower chamber through a connecting tube. The lower chamber is filled with a solution of 1 part glycerine to 3 parts water, which is driven out into the calibrated vessel *C*. When it has risen to the height of the inlet and outlet the flow of gas is interrupted and a portion is trapped. By the time the lower end of the center tube in *C* is sealed off the excess of gas has been forced out against the slight resistance of the elastic bag *P*, and the vessel contains exactly 100 c.c. at atmospheric pressure. During this time the aspirator is drawing the flue gas through the seal *F*. As the liquid rises higher in *C*, the entrapped gas is forced through the small tube *Z* into *A*, which is filled with a solution of caustic potash. Here the CO_2 is absorbed and the potash solution is forced into vessel *B*, which has an outer jacket filled with glycerine supporting a float *N* suspended from the pen-gear *M*. A thin tube through the float keeps the air in *B* at atmospheric pressure. As the float rises the pen-lever swings upward, carrying the pen *Y*, which makes a vertical line upon the clock-driven

FIG. 433. Sarco CO_2 Recorder.

recording drum. The quantity of potash solution displaced, and consequently the lift of the float, is dependent upon the amount of CO_2 absorbed from the 100 c.c. of flue gas. The chart is graduated to indicate directly the percentage absorbed. By the time this operation has been completed water has filled tank K and risen into the siphon G, which, upon starting, very rapidly empties the tank and allows the liquid from C to return to the lower compartment. The float returns to its original position and the remaining gas passes out through E.

Ados CO_2 *Recorder:* Engng., Jan. 12, 1906; Sci. Am. Sup., Dec. 22, 1906.
Sarco CO_2 *Recorder:* U.S. Engr., Nov. 1, 1907, p. 1001.
Uehling Composimeter: Power, June, 1907, p. 404.
American CO_2 *Indicator:* Power, Dec., 1907.
Flue Gas Analysis, Miscellaneous Apparatus: Power, April, 1907, p. 243; Engr. U.S., Jan. 1, 1907, p. 71; Elecn., Lond., Nov. 16, 1906.

410. Moisture in Steam. — Several forms of calorimeters are available for determining the quality of steam. The simplest as well as the most satisfactory, if the percentage of entrained moisture is not beyond its range, is the *throttling* calorimeter, Fig. 435. In this device the sample of steam, which is taken from the steam pipe by means of the perforated nipple, is allowed to expand through a very small orifice into a chamber open to the atmosphere. The excess of heat liberated serves first to evaporate any moisture present and then to superheat the steam at the lower pressure. From the observed temperature and pressures it is easy to calculate, with the aid of steam tables, the percentage of moisture in the original sample.

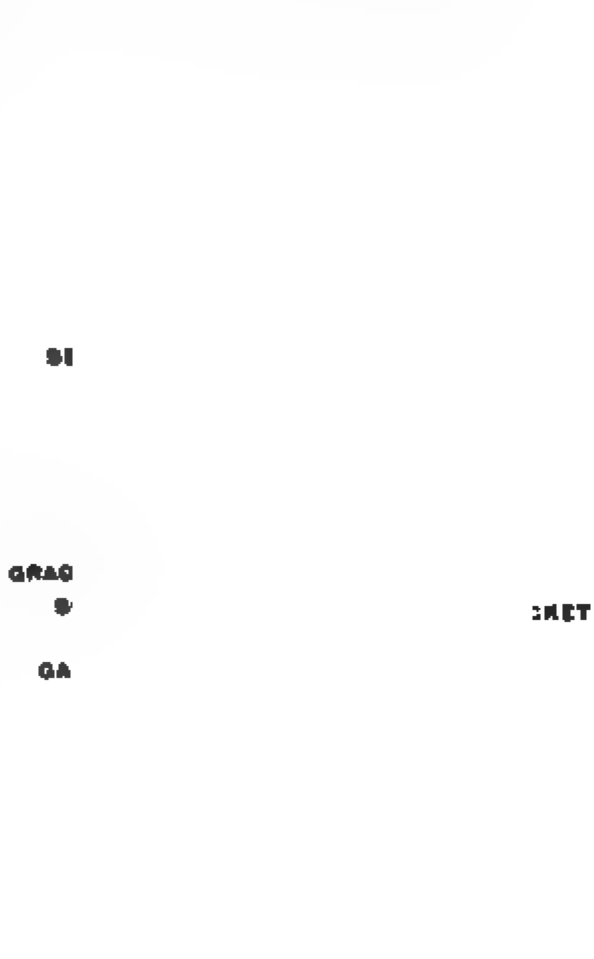

FIG. 434. Carpenter Separating Calorimeter.

The limit of the throttle calorimeter depends upon the steam pressure and is about 3 per cent of moisture at 80 pounds pressure and about 5 per cent at 200 pounds. For steam containing greater percentages of moisture the *separating* calorimeter, Fig. 434, is sometimes used. This instrument is virtually a steam separator and mechanically separates the moisture from the sample of steam.

Fig. 435. A Typical Throttling Calorimeter.

Fig. 435a. Ellison Universal Steam Calorimeter.

The water thus separated collects in a reservoir provided with gauge glass and graduated scale, while the dry steam passes through an orifice to the atmosphere. The weight of dry steam per unit of time is indicated on the gauge, calculated according to Napier's rule, or may be determined by condensing and weighing. The accuracy of the moisture determination is greatly affected by the difficulty of obtaining true samples of steam containing large percentages of moisture.

Fig. 435a shows the Ellison *universal* steam calorimeter, which combines the superheating and throttling principles and is adapted to steam of any degree of wetness. The separating chamber is provided with a gauge glass, not shown, for indicating the weight of water which accumulates only when the steam is too wet to be superheated.

Throttling Calorimeters: Power, Dec., 1907, p. 891; Trans. A.S.M.E., 17-151, 175, 16-448; Engr. U.S., Feb. 15, 1907, p. 219.

Separating Calorimeters: Trans. A.S.M.E., 17-608; Engr. U.S., Feb. 15, 1907, p. 219.

Universal Calorimeter: Trans. A.S.M.E., 11-790.

Thomas Electrical Calorimeter: Power, Nov., 1907, p. 791.

411. Fuel Calorimeters. — The analysis and heat evaluation of fuel require considerable time and skill and much costly apparatus,

FIG. 436. Mahler Bomb Calorimeter.

hence in most power plants it is customary to depend upon a specialist to whom samples are submitted from time to time. In many large

stations, however, the conditions often warrant the establishment of a testing laboratory equipped for the proximate analysis of coal and the determination of the calorific value of the solid, liquid, or gaseous fuel used. The *Mahler bomb* calorimeter illustrated in Fig. 436 is the most accurate and satisfactory device for solid and liquid fuels but is comparatively expensive. The instrument consists of a steel shell or "bomb" of great strength, lined with porcelain or platinum, into which a weighed sample of the fuel is introduced and burned on a platinum pan in the presence of oxygen under a pressure of about 300 pounds per square inch. The charge is ignited by an electric current. During combustion the bomb is submerged in a known weight of water which is kept constantly agitated. The calorific value is calculated from the observed rise in temperature due to the heat evolved, proper corrections being made for the water equivalent of bomb and appurtenances, heat given up by the igniting current, and for radiation or absorption of heat from the surrounding air.

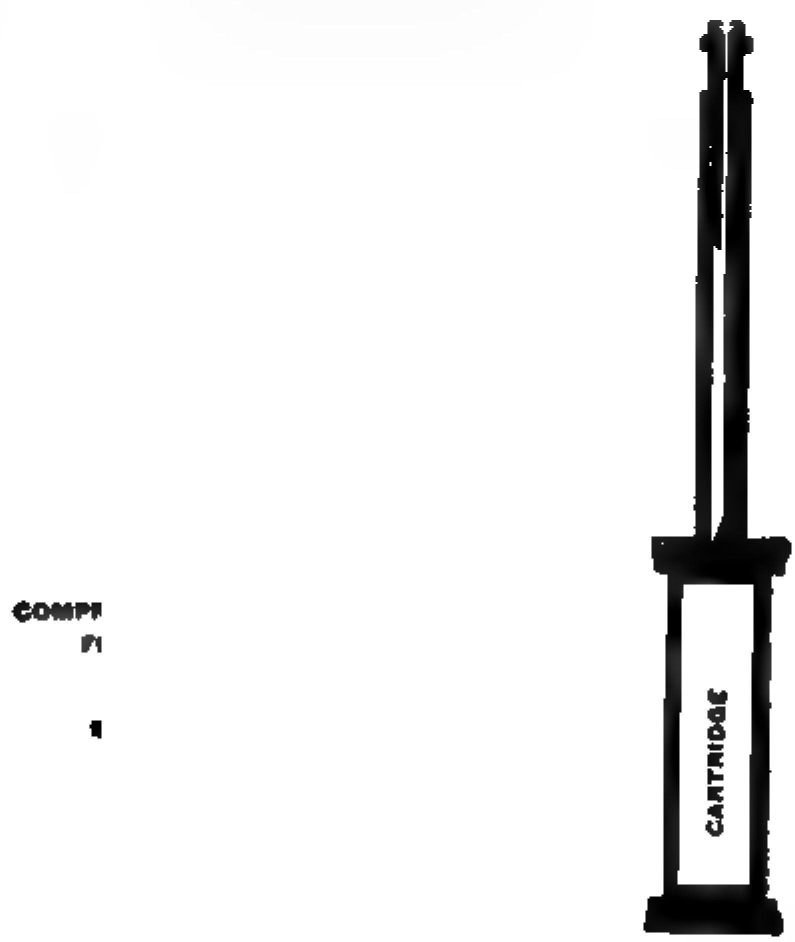

FIG. 437. Parr Fuel Calorimeter.

The *Parr* calorimeter, Fig. 437, is an inexpensive instrument, very simple in operation, and gives results which are sufficiently accurate for all practical purposes. The weighed sample of coal, together with a quantity of sodium peroxide which supplies the oxygen for combustion, is introduced into the cartridge. Means are provided for

rotating the cartridge when submerged in the calorimeter, the attached vanes agitating the water to maintain uniform temperature. The charge is fired either electrically or by introducing a short piece of hot wire through the conical valve. The calorific value is calculated from the observed rise in temperature and the constants of the instrument. Among other forms of instruments, in more or less general use and which give very satisfactory results, may be mentioned the *Carpenter* calorimeter and the *Thompson* calorimeter.

Mahler Bomb Calorimeter: Engr. U.S., Jan. 1, 1907, p. 68.

Parr Fuel Calorimeter: Power, July, 1907, p. 499; Engr. U.S., April 1, 1903; Jour. Am. Chem. Soc., 22–246; The Calorific Value of Fuels, Poole; Gas and Fuel Analysis, Gill; Eng. Chem., Stillman; Chem. Technology, Groves and Trop.

Carpenter Coal Calorimeter: Trans. A.S.M.E., 16–1040.

Thompson Coal Calorimeter: Jour. Soc. Chem. Ind. (1906), 25–409.

Junkers Gas Calorimeter: Jour. Soc. Chem. Ind. (1895), 14–631; Stevens Indicator, Jan., 1905, p. 31.

Comparison of Different Types of Calorimeters: Jour. Soc. Chem. Ind. (1903), 22–1230.

411a. Hamler-Eddy Smoke Recorder. — This apparatus consists essentially of a small motor-driven vacuum pump, which draws a continuous sample of the products of combustion from the uptake breeching or stack and discharges it against a paper-covered drum revolved by clockwork. The *density* of the smoke, the *time* at which visible smoke is being emitted and the *duration* of the smoke production period are automatically recorded on the paper by the smoke itself. Before reaching the pumps the gases pass through a glass " emergency " condenser and a large portion of the vapor content is removed. The pump discharges the partially dried gases against a surface of sulphuric acid (which removes the last trace of moisture) and forces the smoke in the form of a small jet of dry powder onto the surface of the recording paper. The sampling tube leading from the flue to the pump is connected with a steam line and is " blown out " each time a card is changed. The instrument is very compact and portable and may be placed anywhere with respect to the chimney. A number of these appliances in Chicago power plants are giving excellent satisfaction.

CHAPTER XIX.

TYPICAL SPECIFICATIONS.

412. Sample Specifications for a Cross Compound Non-Condensing Engine.—For and in consideration of the amount and terms named in the letter accompanying this specification, and of the same date, we propose to furnish f.o.b. cars at our factory, Elizabethport, N.J., for account of THE ARMOUR INSTITUTE OF TECHNOLOGY, Chicago, Ill., ONE BALL & WOOD HORIZONTAL FOUR-VALVE (CORLISS) CENTER-CRANK ENGINE, designed for direct connection to a direct-current generator, as follows:

General Dimensions.	Horse power	350 to 375
	Diameter of cylinders	H.P., 17; L.P., 27 inches
	Length of stroke	18 inches
	Revolutions per minute	175 to 200
	Governor wheel	diameter, 90; face, 21 inches
	Width of belt (if belted)	20 inches
	Diameter of steam pipe	7 inches
	Diameter of exhaust pipe	10 inches
	Crosshead pins	6 in. long, 8 in. diameter
	Crank pins	9 in. long, $9\frac{1}{4}$ in. diameter
	Main bearings	20 in. long, 9 in. diameter
	Wearing surface of crossheads	242 square inches
	Width of engine over all	14 feet 6 inches
	Length of engine over all	20 feet 3 inches
	Weight complete	60,000 pounds

Rating. The rated power of the engine specified is based on an initial pressure of 120 pounds (measured in the cylinder), cutting off at about one third stroke in both cylinders, without vacuum, when operating at 200 revolutions per minute.

Fittings. With each engine is furnished the following complete list of fittings:

- One extended shaft (omitted if belted engine),
- One sub-base with extension for dynamo (omitted if belted engine),
- One self-oiling outboard bearing (omitted if belted engine),
- One throttle valve,
- One lubricating system consisting of pipes, sight feeds. and oil reservoir,
- One cylinder lubricator, nickel plated,

Fittings — *Continued.*

One set special steel wrenches,
One socket wrench for piston,
One socket wrench for connecting rod bolts,
One steel wrench for hexagon nuts,
One connecting rod set screw wrench,
One monkey wrench,
One spanner for valve stem gland,
One eye bolt for pillow block cap,
Two push-off bolts,
One set grease cups,
Two oil cups,
One hand oil pump,
One set brass oil cups,
Three nipples for drip pipes, for frame,
Four nipples for cylinder drips, 3 inches long,
One nipple for throttle bleeder,
One globe valve for throttle bleeder,
Four globe valves for cylinder drips,
One set foundation bolts, nuts, and plates,
One template for locating bolts,
One governor wheel and keys,
One balance wheel and keys (omitted if direct connected engine),
Packing for piston and valve rods,
One one-gallon can cylinder oil,
One one-gallon can engine oil,
Two cans grease,
Two wedges for wheels,

Cylinders. Cylinders are made of hard and close-grained iron, and under the influence of oil and wear the walls will rapidly acquire a fine, smooth glaze. Radiation is prevented by a thick jacket of asbestos cement, outside of which is neatly fitted an ornamented jacket. Openings and globe valves are provided for drainage.

Connecting Rods. The connecting rods are of forged machinery steel of low carbon and fitted with heavy straps with keys and bolts for adjustment.

Crank Pin Boxes. The crank pin boxes are of cast iron lined with Babbitt metal.

Crosshead Boxes. The crosshead boxes are of the best quality of phosphor bronze.

Crossheads. The crossheads are of cast iron faced with Babbitt metal on the wearing surfaces. The crosshead pins are pressed into the crossheads.

Piston. The piston rods are of special hammered steel, threaded and screwed into the crossheads, and locked fast with special nut counterbored at the end to cover threads, finished and case-hardened. The other ends of the rods will be fitted to the pistons with thread and locked with nut. The pistons will be fitted with two rings turned eccentric and cut open at the thinnest part, the ends being halved so as to lap when in position.

Valves. Both the admission and exhaust valves are of the Corliss pattern. The former are provided with double ports, and are actuated from a wrist plate receiving its motion from the governor placed in the fly wheel of the engine. This governor controls the valves of both the high and low-pressure cylinders and possesses a range of cut-off from 0 to about $\frac{3}{4}$ stroke.

The exhaust valves are driven from a wrist plate through an adjustable eccentric by which any desired degree of compression can be obtained.

Speed. The use of Corliss valves, arranged as described in the foregoing paragraph, permits an increased speed over the common type of Corliss engine with releasing gear, and while yielding the same economy dispenses with many working parts, and, what is more important, with the large and cumbersome fly wheel which has so often proved a source of danger in slow-speed engines.

Frame. The frame is proportioned for great strength and the metal is placed where it is most needed. An oil groove is cast around the bottom of the frame to protect the foundation.

Main Bearings. Main bearings are fitted with removable Babbitt shells which can be replaced when necessary. Special care is taken to have these bearings of ample length to support the wheels and stand the strain of power transmission or the weight of armature when direct connected. In the latter case a self-oiling outboard bearing is provided to carry the outer end of shaft.

Guides. The guides are known as the locomotive pattern and are interchangeable. They are carefully scraped to surface plates and provision made for taking up wear.

Crank Shaft. The crank shaft is of the best quality of steel, being carefully counterbalanced by cast-iron disks in which the necessary weight is placed. In direct connected engines this shaft is either extended in one piece to carry the armature or made in two pieces and coupled.

Governor. The governor is of the inertia type and has a swinging eccentric, the eccentric center moving across the end of the shaft about an outside point, and giving a lead which varies with the point of

Governor — *Continued.*

cut-off from a maximum, at the latest point, to zero, when the governor weights occupy their extreme outward position. Alteration in speed is obtained by changing the amount of weight in the pockets of the lever arm.

Balance. The balance wheel (in the case of belted engine) is made with a flanged rim and with a split hub, the hub being secured to the shaft by a bolt. The other keys are square, with parallel sides, and are inserted without driving.

Oiling. The oiling system consists of a simple oil reservoir which supplies oil through a system of pipes to the points of the engine needing lubrication. After fulfilling its functions this oil is drained and can be used anew. This does away with the old cumbersome oil-cup system and has the great advantage of delivering clean oil to the engine.

Guarantees. *Material.* We guarantee that the material and workmanship are of the best and that all working parts having flat surfaces are scraped to surface plates.

Regulation. That the engine shall regulate within 2 per cent under changes of load within the range of the governor, and that no reduction of boiler pressure shall reduce the speed until the latest point of out-off is reached.

Steam Consumption. That the steam consumption, when the engine is developing its rated power at 125 pounds pressure and no vacuum, shall not exceed 22 pounds of dry steam per indicated horse power per hour; that the clearance shall not exceed 8 per cent.

Drawings. With the engine is furnished a drawing showing its details, together with foundation plans.

Preparation for Shipment. Every engine is completely erected at our works before shipment. The castings are rubbed smooth, carefully filled, and the engine given two good coats of standard shop color. All bright parts are carefully protected against corrosion. The engine is dismantled, the small parts being boxed, and in the case of export shipment the larger pieces crated.

Erection. Full drawings and directions for erecting the engine will be furnished. Template, foundation bolts, nuts, and plates to be shipped in advance if necessary, and by freight unless otherwise directed.

If requested we will furnish the services of an expert to superintend the erection of this engine at the rate of $5 per day added to his traveling expenses and board, the purchaser to furnish all laboring help.

Terms. One-half cash on presentation of bill of lading, balance on completion of erection.

The title to the apparatus herein sold shall not pass from The Ball and Wood Company until all payments hereunder (including deferred payments, if any) shall have been fully made in cash. The purchaser agrees to do all acts necessary to perfect and maintain such retention of title in the said Company. All previous communications between the parties hereto, verbal or written, are hereby abrogated and withdrawn, and this proposal, when duly signed and approved, constitutes the agreement between the parties hereto, and no modification of this accepted agreement shall be binding upon the parties hereto or either of them unless such modification shall be in writing, duly accepted by the purchaser and approved by an executive of the Company.

Limit. Prices subject to revision after thirty days. Delivery subject to strikes, accidents, or causes beyond our control.

413. Specifications for Horizontal Tubular Steam Boiler.* — The following specifications for one 54-inch horizontal return tubular steam boiler, pressure 125 pounds, were prepared by the HARTFORD STEAM BOILER INSPECTION AND INSURANCE COMPANY for the ARMOUR INSTITUTE OF TECHNOLOGY, Chicago.

The boiler to conform to the following conditions and requirements:

Type and General Dimensions. It is to be of the horizontal tubular type, set with overhanging front, and all parts and pieces are to be designed accordingly.

It is to be 17 feet 2 inches long, outside, and 54 inches in diameter, measured on the outside of the smallest ring of plates. Heads are to be 16 feet 0 inches apart, outside.

Materials: Quality, Thickness, and Tests. Shell plates are to be three-eighths of an inch thick on the edges, of open-hearth fire-box steel, having a tensile strength of not less than 55,000 pounds nor more than 62,000 pounds per square inch of section, and an elastic limit of not less than half the tensile strength, with not less than 56 per cent of ductility, as indicated by contraction of area at point of fracture under test, and by an elongation of 25 per cent in a length of 8 inches.

Heads are to be one-half of an inch thick, of best open-hearth flange steel. All plates, both of shell and heads, are to be plainly stamped with name of maker, brand, and tensile strength; brands so located that they may be seen on each plate after the boiler is finished.

Each shell plate is to bear a coupon which shall be sheared off, finished up, and tested by, or for, the maker of the boiler, at his expense. Each coupon is to fulfill the foregoing requirements as

* Drawings have been omitted.

Materials: Quality, Thickness, and Tests—*Continued.*

to strength and ductility, and stand bending down double when cold, when red hot, and after being heated and quenched in cold water, without signs of fracture. There is not to be more than 0.035 per cent of sulphur, nor more than 0.035 per cent of phosphorus in the chemical composition of the plates and heads. All plates failing to pass these tests will be rejected. All tests and inspections of material may be made at the place of manufacture prior to shipment. Certified copies of report of tests must be sent to the Hartford Steam Boiler Inspection and Insurance Company, Hartford, Conn.

Riveting. The longitudinal seams are to be of the double-riveted butt-joint type with double covering strips. They are to be arranged to come well above the fire line of the boiler, and break joints in the 3 ring courses in the usual manner. The plates are to be planed on the caulking edges before rolling.

All dimensions and proportions are to be shown on accompanying drawing No. 1502.

The girth seams are to be of the single-riveted lap-joint type; rivets to be of same size as those in longitudinal seams, and pitched $2\frac{1}{8}$ inches apart from center to center; the distance from center of rivet to the edge of the plate to be equal to $1\frac{1}{2}$ times the diameter of rivet hole.

The rivet holes are to be either drilled in place, or punched at least one-quarter of an inch less than full size; if the latter method is used, the plates, after punching, are to be rolled and bolted together, and the rivet holes drilled in place one-sixteenth of an inch larger than the diameter of the rivets. The plates are then to be disconnected. All burrs are to be removed from the edges of the holes. Should any holes be in the least out of true, they are to be brought in line with a reamer or drill; if a drift-pin is used for this purpose the boiler will be rejected.

All rivets are to be driven by hydraulic pressure, wherever possible, and allowed to cool and shrink under pressure. This pressure is to completely fill the rivet holes, producing a tight joint.

Rivet Hammer Tests. The rivets are to be of the best quality of iron or soft steel, capable of being hammered flat, when cold, to a thickness of one-half their original diameter, or when hot, to one-third their original diameter, without showing signs of fracture. In the absence of physical test, it is understood that the contractor guarantees the above quality of rivets.

Braces. There are to be 20 braces $1\frac{1}{16}$ inches in diameter in the boiler, 10 above the tubes on front head, and 10 on rear head, of the crow-foot form, arranged as shown on drawing. None of them is to be less than 3 feet 6 inches long, and each is to be fastened

Braces — *Continued.*

to shell and heads by two seven-eighths inch rivets at each end; or solid steel, diagonal braces of approved pattern, and of equal strength to the former, may be used. Care is to be exercised in setting them that they may bear uniform tension. Crow-foot braces may be flat in body, if of equal strength to those specified above.

Braces below Tubes. There are to be 4 braces below the tubes in the boiler. Two of these are to be *through* braces extending from head to head. Each brace is to be $1\frac{5}{16}$ inches in diameter, with a fork formed on rear and secured with a $1\frac{1}{8}$-inch turned bolt and nut to a crow-foot securely riveted to rear head; these are the inner or central braces. The front end of brace is to be upset to a diameter of $1\frac{5}{8}$ inches, threaded and secured to front head with a nut and washer on both the inside and outside of head.

The 2 remaining braces are each to be $1\frac{5}{16}$ inches in diameter, and secured to rear head in same manner as the through braces; the front end of the brace is to be extended forward, fitted to side of shell, and riveted there with two 1-inch rivets. All to be substantially as shown on accompanying diagram of tube head No. 2431.

Tubes: Size, Number, and Arrangement. There are to be 36 lap-welded or seamless-drawn tubes, of the best quality with regard to tensile strength and ductility. They are to be round, straight, and free from all surface defects, properly annealed on their ends, and guaranteed by the manufacturers to have been tested to at least five hundred (500) pounds per square inch internal hydrostatic pressure. Each tube is to be 4 inches in diameter, 16 feet 0 inches long, and not less than standard thickness, set in vertical rows, with a clear space between them, vertically and horizontally, of 1 inch, except the central vertical space, which is to be 2 inches, as shown on accompanying diagram of tube head No. 2431.

Holes for tubes are to be neatly chamfered off on outside. Tubes to be set with a Dudgeon expander, and beaded down at each end. Tube holes may be drilled and reamed, or may be punched one-quarter inch less than full size, then rose bitted to exact diameter.

Manholes. There are to be two manholes, one 11 x 15 inches, with pressed steel frame, double riveted to inside of shell on top, and one 10 x 15 inches, flanged in front head below tubes, with suitable plates, yokes, and bolts, the proportions of the whole such as will make them as strong as any portion of the shell of like area.

Boiler Supports. The boiler to be suspended from steel I beams, 6 inches deep, $12\frac{1}{4}$ pounds per foot, by means of eye or U bolts and plate loops. There are to be 6 loops, 2 on each side of the boiler, securely riveted to boiler shell. The I beams are to be supported on cast-

Boiler Supports — *Continued.*

iron columns of square or rectangular section 6 inches square, three-quarters inch thick. Each pair of beams is to be connected together, 3 inches apart, by tie-bolts and cast-iron separators; one separator near each end, and others at intervals of about five feet. The top and bottom flanges of columns are to be faced true.

The whole system of suspension is to be made in the best manner, properly arranged to allow free expansion of the boiler, securely held and supported in every direction, amply strong in every part, and finished complete.

Nozzles. There are to be two heavy cast nozzles, made of gun-iron or steel, one 4 inches internal diameter for steam-pipe connection, and one 6 inches internal diameter for safety-valve connection, each accurately squared on top flange, and securely riveted to boiler on top. Forged or pressed steel pipe flanges may be used in place of nozzles.

The flanges of the nozzles to correspond in diameter and thickness with standard extra heavy pipe fittings.

Smoke Opening. There is to be an opening 10 by 62 inches cut out of front connection on top for attachment of uptake or flue.

Feed Pipe. There is to be a hole tapped in front head for a brass bushing, 3 inches above the top of upper row of tubes, and 16 inches from center of boiler, on left-hand side, for 1¼-inch feed-pipe connection. The bushing is to be not less than 2 inches long, to permit both the external and internal feed pipes to be screwed into it not less than seven-eighths inch.

Also furnish and put in a 1¼-inch feed pipe extending from front head back to within two feet of rear head of boiler, thence across the boiler to near shell on right-hand side. On this end place an elbow with the outlet pointed down as shown on drawings. Feed pipe is to be properly hung from the braces.

Blow-off Pipe Connection. There is to be an extra heavy pressed steel pipe flange, riveted to bottom of shell, near rear end, and tapped to receive a 4-inch extra heavy blow-off pipe. Blow-off valve and fittings to be extra heavy.

Fusible Plug. There is to be a fusible plug in rear head, two inches above top of upper row of tubes.

Fittings. There is to be furnished one pop safety valve 3 inches in diameter, one 6-inch steam gauge, three three-quarters-inch gauge cocks, and one three-quarters-inch gauge glass 12 inches long, all to be of approved pattern, and the necessary holes to be made for their proper connection. If combination water column is used, the steam and water connections between it and the boiler must be made by pipe not less that 1¼ inches in diameter.

Castings for Setting. There is to be furnished a substantial cast-iron front, with all necessary anchor bolts, 10 feet long, closely fitting front connection doors with suitable fastening to prevent warping, closely fitting furnace doors with liner plates, rear connection door 16 x 24 inches, with liner plates, grate bars for grate, pattern to be selected by purchaser of boiler, 54 inches long by 48 inches wide, with suitable bearer bars for same, arch bars for rear connection, and all buckstaves, with the necessary bolts or tie rods, and all other castings or ironwork of any description necessary for the proper construction and setting of the boiler complete.

In General. The intent of the foregoing specification is to provide for material and workmanship of the best quality, and any details of equipment not mentioned in this specification, or not shown on the drawings, but necessary for the proper completion of the boiler ready for operation, and to be hereafter contracted for, must be of equally good quality.

The size and description of parts are to conform substantially to the details of the accompanying plan, and the boiler, complete, is to be delivered at ——— and all of the material and workmanship is to be subjected to the inspection and approval of the Hartford Steam Boiler Inspection and Insurance Company.

414. Specifications for Barometric Condenser and Auxiliaries. — The following sample specifications for a barometric condenser will give some idea of the various items called for in the purchase of a condenser and appurtenances, the italicized items being specified by the purchaser.

"We submit herewith our tender for one ——— ——— condensing plant as follows:

Rated Capacity. The condenser and auxiliary machinery will have sufficient capacity to condense *250* pounds of steam per *minute* (equivalent to the steam exhausted from engines developing *1000* horse power on a basis of *15* pounds of steam per horse power per hour) when supplied with cooling water at a temperature of *70* degrees Fahrenheit, and maintaining a vacuum at the condenser of *26* inches of mercury.

Capacity under Variable Conditions. The plant will also have to condense the quantities of steam under the varying conditions as stated below:

Steam Condensed per Minute, Pounds.	Temperature of Cooling Water, Degrees F.	Vacuum Maintained at Condenser, Inches of Mercury.
250	70	26
300	70	25
340	70	24
250	80	25
290	80	24

Quantity of Cooling Water. The volume of cooling water required when the condenser is working under the above conditions will be from *550* gallons to *650* gallons per minute.

Apparatus Furnished. The apparatus to be furnished by us will consist of:

One cast-iron condensing vessel, complete with barometric tubes and foot valves.

One automatic vacuum regulator.

A structural steel framework for supporting the condensing vessel.

One positive rotary pump for supplying the cooling water to the condenser.

One "dry" air pump.

One horizontal steam engine, arranged to drive the water pump by belt and the air pump direct, the latter placed tandem to the engine; the engine is to be fitted with a suitable governor arranged for variable speeds. Purchaser to furnish belt.

Two pulleys, one for the engine and one for the water pump.

One vacuum gauge.

Four thermometers.

We do not include any steam, air, or water pipes, valves nor foundation bolts, but will furnish plans showing suitable foundations and general arrangement of the machinery.

Price and Delivery. Our price for one barometric condensing plant, as described, including all royalties, and delivered f.o.b. cars our works, is ————————

Terms. Payments as follows: Monthly payments as the work progresses in our shops, less 10 per cent. The retained percentage to be paid when the condenser is started in service, provided this is done within a reasonable time after completion.

Superintendence. If desired, we will furnish a competent machinist to superintend the erection and starting of the plant, charging extra for his services, 50 cents per hour and his traveling and boarding expenses.

Steam Pressure. The engine driving the air and water pumps will be capable of starting and operating the plant with *100* pounds minimum steam pressure, and will be built strong enough to work under *135* pounds maximum steam pressure.

Head Pumped against. The water pump and engine driving same will be designed to raise the injection water from the cold well to the condenser, the level of the water in the cold well to be *not more than 10 feet below* the level of the water in the hot well.

Power Consumption of Auxiliaries. The engine driving the air and water pumps will require approximately 2 per cent of the main engine steam when operating under rated load and with conditions as above stated.

The rotary pump has a positive displacement (Bibus' patent)

Power Consumption of Auxiliaries — *Continued.*

and is of substantial construction. All the power for propelling the water is transmitted through the main shaft, the office of the gears being simply to keep the sealing runner in time with the propelling runner. Stuffing boxes are placed between the pump chamber and bearings to prevent any grit in the water coming into contact with the journals.

The engine and air pump is of the crank and fly wheel type, the engine being fitted with a suitable governor arranged for variable speeds, and the air pump with the ——— patented slide valve.

415. Specification for Steam, Exhaust, Water, and Condenser Piping for an Electric Power Station.* — It is intended that this specification shall cover the complete installation of steam piping, exhaust piping, injection and discharge piping, drain, drip, blow-off, and boiler-feed piping, water piping, valves, separators, anchors, fittings, etc., substantially as shown on the accompanying drawing or hereinafter described.

All the materials used throughout must be the best of their respective kinds, subject to the inspection and approval of the engineer of the purchaser.

The entire work provided for in this specification is to be constructed and finished in every part in a good, substantial, and workmanlike manner, according to the accompanying drawings and this specification to the full intent and meaning of the same, and everything necessary for the proper and complete execution of the plans and drawings, whether the same may have been herein particularly specified or not, or indicated in the plans referred to, to be done and furnished in a manner corresponding with the rest of the work as well, as truly, and as faithfully as if the same were herein particularly described and specifically provided for.

The engineer shall have full power at any time during the progress of the work to reject any materials he may deem unsuitable for the purpose for which they are intended, or which are not in strict conformity with the spirit of this specification. He shall also have the power to cause any inferior or unsafe work to be taken down and altered at the cost of the contractor.

It is to be understood that the final inspection and acceptance of the work are to take place at the building after erection, and that any inspection and acceptance of material and workmanship at the mills, shops, etc., to facilitate the progress of the work, shall not preclude rejection at the building if the same be unsuitable.

* Stevens Indicator, Vol. 18, p. 373, 1901.

Any disagreement or difference between the purchaser and the contractor, upon any matter or thing arising from this specification or the drawings which form a part thereof, shall be referred to the engineer, whose decision and interpretation of the same shall be considered final, conclusive, and binding upon both parties.

Risk and Blame. — The contractor is to assume all risks and bear any loss occasioned by neglect or accident during the progress of the work until the same shall have been completed and accepted by the engineer.

Permits. — The contractor must pay for all permits and inspectors' fees or any other charges from borough, city, county, or state officers.

Dimensions. — During the progress of the work the contractor will be required to keep at the building site a complete set of drawings and a copy of this specification. The contractor will consider dimensions shown on drawings to be approximate as to location of machinery, but sufficiently accurate for the purpose. Absolute dimensions must be gotten from the location of the boilers, engines, and other machinery after they are set. No drawing shall be scaled.

The purchaser reserves the right to put other parties at work on the premises erecting machinery and doing other work during the continuance of this contract.

The contractor must conform to such rules and regulations as are in force on the property and carry out this contract with as little interference as possible with the other work of the purchaser.

Drawings.*

The following is a list of the drawings which accompany this specification and which form a part thereof:

(Title) .... No. (Drawing Number).........
do. .... do. .... etc.

Description of Plant.

Steam Piping. — The plant consists of four (4) 250-horse-power boilers from which steam will be led to two (2) independent 16-inch steam headers. Two (2) 8-inch take-offs will be carried from each boiler, one (1) to each header. These 8-inch take-offs will consist of large radius wrought-iron bends of a quality hereinafter described. From each header a 6-inch supply will be led to the low-pressure cylinders of these engines. From the header nearest the engine room two (2) 6-inch supplies will be led through the partition wall into the

* Drawings have been omitted.

cellar for the purpose of furnishing steam to the condenser pumps. From one (1) header a 6-inch and 5-inch supply will be carried to a tandem compound engine and thence 5-inch to the two (2) exciter engines.

The condenser steam lines will be continued under the engine-room floor so as to form a reserve steam supply to the exciter engines. From the end of each 16-inch main steam header a 3-inch loop will be carried to and from the fire and boiler feed pumps. The general arrangement with sizes of pipes, separators, valves, etc., is shown on drawing No. ——.

The contractor's high-pressure steam work will begin with the steam nozzles on the boilers and end with the throttle valves of pumps and engines. The contractor will furnish all pump throttle valves but not engine throttle valves.

Condenser and Exhaust Piping. — The three (3) jet condensers and air pumps will be set under engine-room floor. Injection water will be led to the condenser through a 16-inch cast-iron pipe which begins at a point 6 feet outside the wall of the boiler house and runs from this point to the condensers. After leaving the connections for the fourth condenser, this main reduces to 14 inches in diameter; after leaving the third it reduces to 10 inches in diameter; and after leaving the second it reduces to 3 inches in diameter.

This injection water main will be connected with the city supply by a 5-inch cast-iron water pipe; and the discharge pipe will be of cast-iron and will be led from the condensers, beginning with a diameter of 5 inches, increasing to 10 inches, and again increasing to 14 inches, and then to 16 inches after the last condenser, from whence it will be carried to a point 6 feet outside of the boiler-house wall.

The low-pressure cylinder of the tandem compound engine will be connected by a 10-inch and a 12-inch pipe with No. 1 condenser. Connection will be made between both high- and low-pressure cylinders of the 750-horse-power engines with 18-inch condenser exhaust headers. Between the exhaust header and the cylinder of the engines, connections will be made with a free exhaust main leading to and from exhaust riser extending through the roof of the boiler house. The general arrangement and the sizes of the gate valves, free exhaust valves, etc., are shown on drawing No. ——.

The exhaust from the two (2) exciter engines and the three (3) condensers, the boiler feed pumps, and the fire pumps will be collected and led to a 1500-horse-power open-type feed-water heater located in the boiler room. This heater will be provided with an exhaust riser and exhaust head extending through the roof of the boiler house.

The arrangement of the auxiliary exhaust pipe is shown on drawing No.——.

Boiler Feed Water Piping. — To the feed-water heater above mentioned a 2½-inch supply of fresh water will be furnished. The boiler feed pumps will draw the feed water from the heater through a 6-inch cast-iron suction pipe. They will deliver it through 3-inch brass pipes to two (2) hot-water meters equipped with by-passes and thence through a 3-inch brass pipe line to boilers. A reserve wrought-iron 3-inch feed pipe line will be run parallel to the brass feed pipe line and so connected up to it that it may be used as a reserve in case of accident to the brass line. Connections from 3-inch lines to boilers will be 2½-inch brass pipe.

The general arrangement of the feed pipe, including gate valves, check valves, sizes, etc., is shown on drawing No. ——.

Drip Piping. — There will be two (2) systems of drip piping, including traps, trap by-passes, valves, etc.

The high-pressure system comprises all the drips from the main steam header, separators, and high-pressure steam lines. The water from these traps will be led to the feed-water heater, drawing No. ——.

The low-pressure system comprises all the drips from exhaust mains, receivers, and cylinder drain cocks.

This water will be led to a catch-basin from whence it will be pumped by a tank pump. This pump will be furnished by the purchaser, drawing No. ——.

Blow-off Pipes. — A separate and distinct wrought-iron, 2½-inch blow-off pipe will be run from the blow-off valve of each of the four (4) water-tube boilers. These pipes will be run in trenches in the boiler-room floor to catch-basins located outside of building. The purchaser will construct the trenches and catch-basins.

Water Piping. — The purchaser will make a 6-inch connection from the city main to the northwest corner of the boiler room. From this point the contractor will run 6-inch cast-iron pipe to the inlet of the fire pump. From this he will run 6-inch pipe to the several fire hydrants. He will connect this 6-inch line with the injection water main with a 5-inch cast-iron connection. Near the fire pump he will run a 2½-inch connection to the feed-water heater. The pipe around the fire pump will be so arranged that ordinarily city pressure will be maintained on the water system, but in case of need the fire pump may be used to increase this pressure.

This piping is all shown on drawing No. ——.

Wrought-Iron Pipe. — All wrought-iron pipes or steam lines to be

guaranteed full-weight lap-welded wrought-iron pipe, in accordance with the standard dimensions as given in the table below. But the 16-inch headers will be 16 inches O.D., 15.04 inches I.D., and shall weigh 87 pounds per foot.

Nominal Internal.	Actual External.	Actual Internal.	Nominal Weight per Foot.
Inches.	Inches.	Inches.	Pounds.
1	1.315	1.048	1.663
$1\frac{1}{4}$	1.66	1.38	2.444
$1\frac{1}{2}$	1.9	1.611	2.678
2	2.375	2.067	3.609
$2\frac{1}{2}$	2.875	2.468	5.739
3	3.5	3.067	7.536
$3\frac{1}{2}$	4	3.548	9.001
4	4.5	4.026	10.665
$4\frac{1}{2}$	5	4.508	12.34
5	5.563	5.045	14.502
6	6.624	6.065	18.762
8	8.625	7.982	28.35
10	10.75	10.019	40.065
14	15	14.25	57.893
15	16	15.25	71.77

All pipes carrying hot water, such as feed-water, blow-offs, and high-pressure drip pipes, are to be extra heavy.

All lap-welded pipe shall be proved to 500 pounds pressure per square inch before shipment. Butt-welded pipe shall be proved to 300 pounds pressure per square inch before shipment.

Exhaust pipes are to be wrought iron or steel in every case; sizes 8 inches and above may be light O.D. tubing with flanges peened or expanded on, but must be absolutely tight at 28 inches vacuum. These pipes must be tested to 25 pounds hydrostatic pressure after erection.

All high-pressure wrought-iron pipe when in place is to be tested at 250 pounds air pressure, and must be good for a working pressure of 175 pounds. All high-pressure steam pipe to be tested after erection at 175 pounds steam pressure, to the satisfaction of the engineer in every particular.

Cast-Iron Pipe. — The cast-iron pipe for injection and discharge water and for all other purposes, as shown on the accompanying drawings, shall be made of tough gray iron not less than $\frac{5}{8}$ inch thick at any point, free from blowholes, true to pattern, and of workman-like finish. It shall be tested at 250 pounds pressure before erection, and when in place shall be tested to 25 pounds hydrostatic pressure.

Cast-iron pipes inside of buildings are to be flanged with flanged fittings.

Outside of buildings they shall be bell and spigot ends, and must be coated with tar both inside and outside.

Valves.— All steam valves over two inches, except throttle valves, will be gate valves. All gate valves will be Chapman's best make or equivalent.

On high-pressure steam lines, valves must be fitted with bronze removable seats, outside screw and yoke, and by-passed from 5 inches up.

On exhaust lines, standard pattern gate valves will be used (Chapman or equivalent), with inside screw.

Globe valves on high-pressure steam and water lines will be Schutte's make or equivalent.

Free exhaust valves will be either Schutte's make or equivalent.

Exhaust valves between engines and condensers, injection water valves at condensers, as well as priming water valves, will have their stems extended and fitted with floor stands, so that they can be operated from floor above. Floor stands will be polished all over, with polished hand wheels and indicators.

All high-pressure valves will be tested to the satisfaction of the engineer, at a hydrostatic pressure of 350 pounds per square inch and air pressure of 100 pounds between gates.

Standard valves for exhaust and water will be tested to 150 pounds.

Flanges and Fittings. — All flanges and fittings for high-pressure steam lines 3 inches and over will be open-hearth steel castings. These castings must be perfectly solid, made from heavy patterns, and free from blowholes or other defects. They must be tested at works to 500 pounds hydrostatic pressure, and guaranteed for a working steam pressure of 175 pounds per square inch.

The diameter, thickness, and drilling of flanges must not be less than the dimensions given in the following table.

Size of Pipe.	Diameter of Flange.	Thickness.	Number of Bolts.	Size of Bolts.
Inches.	Inches.	Inches.		Inch.
16	25	$1\frac{3}{4}$	20	1
14	23	$1\frac{5}{8}$	16	1
12	20	$1\frac{1}{2}$	16	1
10	$17\frac{1}{2}$	$1\frac{3}{8}$	12	1
9	16	$1\frac{5}{16}$	12	$\frac{7}{8}$
8	15	$1\frac{1}{4}$	12	$\frac{7}{8}$
7	14	$1\frac{1}{4}$	12	$\frac{3}{4}$
6	13	$1\frac{3}{16}$	8	$\frac{7}{8}$
5	11	$1\frac{1}{8}$	8	$\frac{3}{4}$
$4\frac{1}{2}$	$10\frac{1}{2}$	$1\frac{1}{16}$	8	$\frac{3}{4}$
4	10	$1\frac{1}{16}$	8	$\frac{3}{4}$
$3\frac{1}{2}$	9	1	8	$\frac{5}{8}$
3	9	1	8	$\frac{5}{8}$

Below 3 inches, on all high-pressure steam, water, or drip lines, extra heavy screwed cast-iron fittings are to be used, with sufficient extra heavy flange unions, so that any section of pipe can be readily taken out without disturbing the balance.

The brass boiler feed piping will be made up with extra heavy brass screwed fittings made from cast-iron patterns.

Pipe will be iron pipe size.

Brass flange unions will be made from standard cast-iron patterns, and a sufficient number used to make up the sections readily.

On low-pressure piping, standard fittings and flanges are to be used. Sizes 6 inches and above are to be flanged, and below 6 inches, screwed, with sufficient number of flange unions in same.

All high-pressure flanges are to be recessed on face, pipe is to be screwed, then peened up tight, and must stand the tests given above.

Standard flanges must stand the tests as stated.

All bolt holes must be drilled in solid metal; no cored holes will be allowed.

On blow-off lines, long sweep fittings are to be used, with extra heavy flange unions.

Gaskets and Bolts. — All gaskets on high-pressure steam lines must be copper.

On exhaust and water lines metallic gaskets are to be used.

All bolts must have hexagonal heads and nuts, no matter what size, points to be finished and nuts to be cold punched.

Pipe Covering. — All high-pressure steam lines, separators, and all drip lines between high-pressure steam lines and their traps will be covered with an approved magnesia pipe covering. All fittings will have molded magnesia coverings to fit them neatly.

Exhaust Heads. — The contractor will furnish and erect two (2) exhaust heads, one on the free exhaust from the engines and the other on the free exhaust from the feed-water heater.

Pipe Supports. — There will be furnished and erected the necessary wrought-iron hangers to support the two (2) 16-inch headers in the boiler room. There will also be supplied the necessary supports for the steam mains to the pumps and exciter engines. The injection water main and the condenser exhaust header will be hung from the engine room floor beams. The free exhaust header and the discharge water main will be supported on piers and saddles from the cellar floor. The saddles to be furnished by the contractor.

Separators. — The contractor will furnish and erect, as shown on drawing No. ——, eleven (11) steam separators.

Anchors. — All live-steam mains, and especially the 16-inch steam headers in the boiler room, will be anchored so as to secure no movement (at point of anchorage) on account of the expansion and contraction or vibration. Anchors to be placed where needed after the rest of the work is completed.

Relief Valves. — The boiler-feed and fire pumps shall be fitted with automatic controlling devices so that a certain definite maximum pressure may be maintained on the feed-water and fire systems.

Overflow Pipes. — The contractor will furnish an overflow relief valve for the fire pump and will connect same with the drain outside the boiler house. He will also connect the blow-off and overflow from the feed-water heater with the said drain.

Long Radius Bends. — Where shown on the drawings, changes in direction of pipe runs will be made with long radius bends. These bends will be of wrought-iron of the quality described above. Nine inches of straight pipe will be left on the ends of the bends to cut necessary threads.

The following table shows the minimum radii for these wrought-iron bends.

Size.	Minimum Radius.	Size.	Maximum Radius.	
Inches.	Inches.	Inches.	Feet.	Inches.
2	4½	7	3	0
2½	6	8	3	4
3	8	9	4	0
3½	10	10	4	4
4	14	12	5	6
4½	16	14	7	0
5	20	16	7	6
6	24	....	....	

I-Beam Supports. — The contractor will furnish and erect on the top chord of the roof trusses in the boiler room a sufficient length of I beams or channels from which he will hang the 16-inch headers. The steam connections between 16-inch header and the engines will be supported by underneath rod trusses, and vibration will be taken up by means of lateral ties.

Flange Covering. — After all the other work is completed and the plant has been in operation not less than two (2) weeks, the contractor will cover all flanges and other joints with an approved magnesia covering.

Reducing and Relief Valves. — The contractor will furnish and erect two (2) reducing valves on the throttle valves of the low-pressure cylinders of the 750-horse-power engines. These valves will be of the Kiely type or equivalent.

There will also be furnished and erected the necessary traps from the receivers between the high and low-pressure cylinders.

Check Valves. — All drip lines will be equipped with check valves which will be set at the lowest point in the line. Two (2) three-inch brass checks will be placed on the boiler feed pump outlets, all as shown on drawing No.——.

The outlet of the boiler feed pumps will also have a relief valve which will return the water to the feed-water heater in case the pressure on the pump outlet should rise above a certain definite point.

Traps. — All traps on high-pressure drip lines will be extra heavy steam traps.

Painting. — All pipes shall be painted with two (2) coats of slate graphite and linseed oil, or other good pipe paint satisfactory to the engineer.

The exhaust and condenser piping must be thoroughly painted twice under vacuum.

Damper Regulator. — The purchaser will furnish the damper regulator, but the contractor will set it and connect it with both steam headers, water supply, and damper.

Shields. — Where steam mains pass through the partition wall of the building the contractor will neatly close the opening with a sheet-metal shield so as to prevent dust from the boiler room from entering the engine room.

Meters. — The contractor will furnish and connect as shown on drawing No. —— two (2) 3-inch hot-water meters, approved by the engineer, to have a capacity of 200 gallons per minute.

Gauges. — The purchaser will furnish the customary gauges and boards, which will be set up and connected by the contractor.

Machinery. — The purchaser will furnish all engines, pumps, condensers, feed-water heater, boilers, foundations, but the contractor shall connect up the above machinery according to the evident intent and meaning of this specification.

The purchaser will not furnish any pipe, valves, fittings, pipe covering, etc., or anything connected with the work except the machinery mentioned above.

416. Government Specification and Proposal for Supplying Coal.

U. S. TREASURY DEPARTMENT.

United States.................................

..................................., 190..

PROPOSAL.

Sealed proposals will be received at this office until 2 o'clock p. m., ..., 190.., for supplying coal to the United States ....building at ....as follows:

....

....

....

The quantity of coal stated above is based upon the previous annual consumption, and proposals must be made upon the basis of a delivery of 10 per cent more or less than this amount, subject to the actual requirements of the service

Proposals must be made on this form, and include all expenses incident to the delivery and stowage of the coal, which must be delivered in such quantities, and at such times within the fiscal year ending June 30, 190 , as may be required.

Proposals must be accompanied by a deposit (certified check, when practicable, in favor of ....) amounting to 10 per cent of the aggregate amount of the bid submitted, as a guaranty that it is bona fide. Deposits will be returned to unsuccessful bidders immediately after award has been made, but the deposit of the successful bidder will be retained until after the coal shall have been delivered, and final settlement made therefor, as security for the faithful performance of the terms of the contract, with the understanding that the whole or a part thereof may be used to liquidate the value of any deficiencies in quality or delivery that may arise under the terms of the contract.

When the amount of the contract exceeds $10,000, a bond may be executed in the sum of 25 per cent of the contract amount, and in this case, the deposit or certified check submitted with the proposal will be returned after approval of the bond.

The bids will be opened in the presence of the bidders, their representatives, or such of them as may attend, at the time and place above specified.

In determining the award of the contract, consideration will be given to the quality of the coal offered by the bidder, as well as the price per ton, and should it appear to be to the best interests of the Government to award the contract for supplying coal at a price higher than that named in lower bid or bids received, the award will be so made.

The right to reject any or all bids and to waive defects is expressly reserved by the Government.

DESCRIPTION OF COAL DESIRED.*

Bids are desired on coal described as follows:

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Coals containing more than the following percentages, based upon dry coal, will not be considered:

Ash.. ......per cent.

Volatile matter.................................. ......per cent.

Sulphur.. ......per cent.

† Dust and fine coal as delivered at point of consumption ......per cent.

DELIVERY.

The coal shall be delivered in such quantities and at such times as the Government may direct.

In this connection, it may be stated that all the available storage capacity of the coal bunkers will be placed at the disposal of the contractor to facilitate delivery of coal under favorable conditions.

After verbal or written notice has been given to deliver coal under this contract, a further notice may be served in writing upon the contractor to make delivery of the coal so ordered within twenty-four hours after receipt of said second notice.

Should the contractor, for any reason, fail to comply with the second request the Government will be at liberty to buy coal in the open market, and to charge against the contractor any excess in price of coal so purchased over the contract price.

SAMPLING.

Samples of the coal delivered will be taken by a representative of the Government.

In all cases where it is practicable, the coal will be sampled at the time

* NOTE.—This information will be given by the Government as may be determined by boiler and furnace equipment, operating conditions, and the local market.

† NOTE.—All coal which will pass through a ¼-inch round-hole screen.

it is being delivered to the building. In case of small deliveries, it may be necessary to take these samples from the yards or bins. The sample taken will in no case be less than the total of one hundred (100) pounds, to be selected proportionally from the lumps and fine coal, in order that it will in every respect truly represent the quality of coal under consideration.

In order to minimize the loss in the original moisture content the gross sample will be pulverized as rapidly as possible until none of the fragments exceed ½ inch in diameter. The fine coal will then be mixed thoroughly and divided into four equal parts. Opposite quarters will be thrown out, and the remaining portions thoroughly mixed and again quartered, throwing out opposite quarters as before. This process will be continued as rapidly as possible until the final sample is reduced to such amount that all of the final sample thus obtained will be contained in the shipping can or jar and sealed air-tight.

The sample will then be forwarded to the Chief Clerk of the Treasury Department, care of the storekeeper.

If desired by the coal contractor, permission will be given to him, or his representative, to be present and witness the quartering and preparation of the final sample to be forwarded to the Government laboratories.

Immediately on receipt of the sample, it will be analyzed and tested by the Government, following the method adopted by the American Chemical Society, and using a bomb calorimeter. A copy of the result will be mailed to the contractor upon the completion thereof.

CAUSES FOR REJECTION.

A contract entered into under the terms of this specification shall not be binding if, as the result of a practical service test of reasonable duration, the coal fails to give satisfactory results due to excessive clinkering, or to a prohibitive amount of smoke.

It is understood that the coal delivered during the year will be of the same character as that specified by the contractor. It should, therefore, be supplied, as nearly as possible, from the same mine or group of mines.

Coal containing percentages of volatile matter, sulphur, and dust higher than the limits indicated on line 54, and coal containing a percentage of ash in excess of the maximum limits indicated in the following table will be subject to rejection.

In the case of coal which has been delivered and used for trial, or which has been consumed or remains on the premises at the time of the determination of its quality, payment will be made therefor at a reduced price computed under the terms of this specification.

Occasional deliveries containing ash up to the percentage indicated in the column of "Maximum limits for ash," on page 700, may be accepted.

Frequent or continued failure to maintain the standard established by the contractor, however, will be considered sufficient cause for cancellation of the contract.

* PRICE AND PAYMENT.

Payment will be made on the basis of the price named in the proposal for the coal specified therein, corrected for variations in heating value and ash, as shown by analysis, above and below the standard established by contractor in this proposal. For example, if the coal contains two (2) per cent, more or less, British thermal units than the established standard, the price will be increased or decreased two (2) per cent accordingly.

The price will also be further corrected for the percentages of ash. For all coal which by analysis contains less ash than that established in this proposal a premium of 1 cent per ton for each whole per cent less ash will be paid. An increase in the ash content of two (2) per cent over the standard established by contractor will be tolerated without exacting a penalty for the excess of ash. When such excess exceeds two (2) per cent above the standard established, deductions will be made from price paid per ton in accordance with following table:

Ash as established in proposal.	No deduction for limits below.	Cents per ton to be deducted.							Maximum limits for ash.
		2	4	7	12	18	25	35	
		Percentages of ash in dry coal.							
Per cent.									
5	7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	12
6	8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	13
7	9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	14
8	10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	14
9	11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	15
10	12	12–13	13–14	14–15	15–16	16–17	17–18	....	16
11	13	13–14	14–15	15–16	16–17	17–18	18–19	....	16
12	14	14–15	15–16	16–17	17–18	18–19	19–20	....	17
13	15	15–16	16–17	17–18	18–19	19–20	20–21	....	18
14	16	16–17	17–18	18–19	19–20	20–21	21–22	....	19
15	17	17–18	18–19	19–20	20–21	21–22	....	....	19
16	18	18–19	19–20	20–21	21–22	22–23	....	....	20
17	19	19–20	20–21	21–22	22–23	....	....	....	21
18	20	20–21	21–22	22–23	....	....	....	....	22

* NOTE. — The economic value of a fuel is affected by the actual amount of combustible matter it contains, as determined by its heating value shown in British thermal units per pound of fuel, and also by other factors, among which is its ash content. The ash content not only lowers the heating value and decreases the capacity of the furnace, but also materially increases the cost of handling the coal, the labor of firing, and the cost of the removal of ashes, etc.

Proposals to receive consideration must be submitted upon this form and contain all of the information requested.

....

.................................., 190

The undersigned hereby agree to furnish to the U. S........................ building at.................................., the coal described, in tons of 2,240 pounds each and in quantity, 10 per cent more or less than that stated on page 697, as may be required during the fiscal year ending June 30, 190 , in strict accordance with this specification; the coal to be delivered in such quantities and at such times as the Government may direct.

Price per ton (2,240 pounds)..........................$.................

Commercial name of the coal ....

Name of the mine or mines...

Location of the mine or mines...

Name or other designation of the coal bed or vein.........................

Size (indicate information which will apply) —

Unsized............ Lump............ Run of mine.................

Screened, through.....inch and over.....inch {Round, Square} Openings. / Bar screen.

Data to establish a basis for payment:

British thermal units in coal as delivered..............................

Ash in dry coal (Method of American Chemical Society).............per cent.

It is important that the above information does not establish a higher standard than can be actually maintained under the terms of the contact; and in this connection it should be noted that the small samples taken from the mine are invariably of higher quality than the coal actually delivered therefrom. It is evident, therefore, that it will be to the best interests of the contractor to furnish a correct description with average values of the coal offered, as a failure to maintain the standard established by contractor will result in deductions from the contract price, and may cause a cancellation of the contract, while deliveries of a coal of higher grade than quoted will be paid for at an increased price.

Signature:...............................

Address:...............................

Name of corporation, ...

Name of president, ...

Name of secretary, ...

Under what law (State) corporation is organized:..........................

CHAPTER XX.

A TYPICAL CENTRAL STATION.

Steam Turbines. — Alternating Currents.

The Fisk Street Station of the Commonwealth Edison Company, Chicago, is an excellent example of modern central station practice. The present (June, 1910) rated capacity of the plant is 120,000 kilowatts, though space is available for a considerable increase. The station is located on the banks of the Chicago River near Fisk and Twenty-second streets, as indicated in Fig. 438, and is about one and one-half miles south of the center of distribution of the present load. The location of the station between the east and west slips of the river secures an unusual advantage in the location of the intake and discharge tunnels, and the extent of property affords ample storage capacity for coal. Both the Chicago & Alton and the Chicago, Burlington & Quincy railways extend into the property, giving excellent facilities for the transportation of coal, ashes, construction materials, and machinery. The plant is constructed on the unit basis, each turbine and generator having its own boilers, auxiliaries, and piping system, thus permitting any unit to be shut down without interfering with the operation of the rest of the system.

Building. — The main building rests on piles, driven to hard pan, capped with a grillage of I beams and concrete. The walls are of red pressed brick trimmed with white Bedford stone. The windows are 25 feet wide and 32 feet high, the sections of which are operated by compressed air. Fig. 439 gives a view of the north elevation. Large skylights afford ample light and ventilation. The entire interior wall surface of the turbine room is finished with white enameled brick trimmed with terra cotta. The boiler-room walls have an eight-foot wainscoting of enameled brick, the remainder being red pressed brick. The floors are of concrete, that in the turbine room being covered with two-inch hexagonal terra-cotta tile. The roofs are of Roebling concrete. The total width of the building is 243 feet, the turbine room taking up 61 feet, the boiler room 142 feet, and a car track the remainder. A 50-ton motor-driven crane spans the turbine room and is used in connection with units 1 to 4, and a 60-ton crane is provided for the

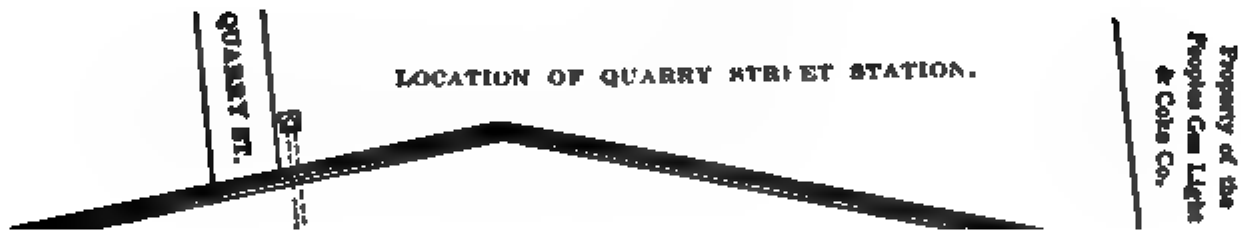

Fig. 438. Fisk Street Station of the Commonwealth-Edison Company, Chicago, Ill.

remaining units. A 5-ton auxiliary hoist is also provided on the main cranes. In the boiler room a small hand-power crane serves each two batteries of boilers.

Fig. 439. North Elevation of Building.

Coal and Ash Handling.— An interior shed extends the entire length of the east end of the building, as indicated in Figs. 440 and 441. Coal is brought in on cars and dumped or shoveled into a track hopper, from which it is delivered to the overhead bunkers by the conveying system. A crusher is placed between the track hopper and main conveyor to be used in case lump coal is furnished. These bunkers have a capacity of 1200 tons each, sufficient for several days run. The conveyors are driven by a 15-horse-power motor and are of the McCaslin pattern, endless chain, with overlapping buckets, each bucket having a capacity of 100 pounds. The conveyors move at a rate of 50 feet per minute, giving a service capacity of 75 tons per hour for each unit. A separate conveyor and bunker is installed in each section of 16 boilers. The coal bunkers feed through flexible down spouts to the stoker magazines. Underneath the front end of the stoker is a fine-coal hopper which collects the fine coal falling through the grate and discharges it into the conveyor system, as in Fig. 99. The ashes collect in the ash pit, from which they are dumped into the conveyor

FIG. 440. General Plan of Units 6 to 10, Fisk Street Station.

FIG. 441. Side Elevation Unit No. 10, Fisk Street Station.

and carried to an ash bin directly over the coal track. Illinois screenings furnish the greater part of the fuel. Provision is also made for outside storage.

Boilers. — The boiler plant is divided into five sections, each section consisting of sixteen 500-horse-power B. & W. boilers arranged in batteries of eight and equipped with B. & W. chain grates. The settings are installed back to back, as illustrated in Fig. 440. Each boiler has two 42-inch steam drums, approximately 5000 square feet of heating surface, and about 1000 square feet of superheating surface. Steam is generated at a pressure of 200 pounds per square inch with superheat of 150 degrees F. The ratio of water-heating surface to grate surface is approximately 55 to 1, and the ratio of the total heating surface to grate surface is about 66 to 1. When burning Illinois screenings, an average depth of 7 inches is maintained on the grate, with speed of grate of 5 inches per minute. The grates are driven by Krehbiel oscillating engines, two engines being provided for each section but only one being in use at a time. The boilers are supported by reënforced girders of the main building structure. A gallery is placed in front of the settings, 8 feet above the floor, to facilitate cleaning of tubes. Galleries are also placed between the batteries and on top of them. Spaces of 5 feet are provided between the sides and rears of the batteries, and 18 feet 8 inches in front. The furnaces are similar to the one illustrated in Fig. 62. The outside of the setting is finished with red pressed brick. Each drum is fitted with a four-inch pop safety valve. The blow-off main is 4 inches in diameter and discharges into the river. There are four blow-off connections to each boiler, each being provided with a blow-off cock and an angle valve; three of the connections are fitted to the mud drum and the other to the superheater drain.

Chimneys. — One stack is provided for each section of 16 boilers. The shaft is supported by the steel work of the boiler setting, as shown in Fig. 442, an arrangement which commends itself where space is limited and real estate values are high. The stacks for all units are 259 feet 6 inches in height above the grate surface, and are 18 feet in internal diameter. The lining is of radial fire brick and varies from 4 inches to 13 inches in thickness. The steel sections are 5 feet high and vary in thickness from $\frac{3}{8}$ inch to $\frac{1}{2}$ inch. There are two flues, one 32 feet long and the other 63 feet, which enter the stack on opposite sides.

Turbines. — The prime movers are vertical five-stage Curtis turbines with base condenser and are rated at 12,000 kilowatts each. The normal speed is 750 r.p.m. The average steam consumption, including all

FIG. 442. Section through Boiler Room.

auxiliaries is approximately 15 pounds per kilowatt hour, corresponding to a coal consumption of 3 pounds per kilowatt hour (Illinois screenings, 10,400 B.T.U. per pound). Special tests have shown as low as 12.8 pounds per kilowatt hour, initial pressure 200 pounds gauge, 150 degrees superheat, absolute back pressure ½ inch of mercury. Each pair of units has a pair of duplicate pumps, an accumulator and a storage tank for supplying oil, the step-bearing pressure being maintained at 750 pounds per square inch. When the accumulator falls below a certain point a motor-driven pump is automatically started.

Generators. — The generators are 2300-volt, 25-cycle, three-phase General Electric alternators mounted over the vertical shaft as illustrated in Fig. 185. Exciting current is furnished by

2 50-kilowatt motor-driven generators.
2 75-kilowatt motor-driven generators.
2 150-kilowatt motor-driven generators.
2 75-kilowatt steam-driven generators.
2 150-kilowatt steam-driven generators.

Part are held in reserve, though no particular units are maintained for the purpose. The high-tension cables lead from the generator through an underground tunnel to the switch house, located about 50 feet west of the main building. The oil switches, wattmeters and other instruments are located on the first floor, while the bus-bars and other high-tension connections are in the basement. The station switchboard or operating gallery in the main building is equipped with only such devices as are necessary for the control of the machines, all other instruments being located in the switch house. From the switch house the high-tension current is conducted through oil switches to the various substations, where it is converted to direct current by rotary converters, or transformed from 25 to 60 cycles by motor generator sets.

The Twenty-second Street substation is located at the north end of the property (Fig. 438). In this substation are installed two motor generator sets and one rotary converter, the latter supplying direct current to the neighboring district and to the main station.

Boiler and Turbine Piping. — Immediately below each boiler section is an apartment called the "header room," where the steam pipes from the various boilers join the main header, which increases in size from 6 inches at the most remote boiler to 10 inches at the middle boiler, and finally to 14 inches where it leaves the nearest boiler and passes to the turbines. The pipes are of wrought iron, with welded flanges, and are packed with copper gaskets. The feeder from each boiler is 6 inches in diameter. An angle stop valve and a check valve are placed at the

boiler nozzle and a globe valve at the header. A motor-operated throttle valve and strainer are placed at the turbine, and a hydraulically oper ated valve, controlled from the operating gallery, is located in the header room. The main header is not anchored at any point, the entire weight being carried by roller supports. The only drain in the header is a ¾-inch bleeder on the turbine side of the hydraulically operated valve. The bleeder is connected to a trap which discharges into either boiler or superheater blow-off main. All branches or feeders are drained and discharged into the superheater blow-off.

Condensers and Auxiliaries. — Each unit has its own condensing apparatus, feed-water heater, hot well and feed pumps. The condensers are of the Worthington "base" type with 25,000 sq. ft. of cooling surface each, composed of 5900–6000 1-inch tubes 16 feet long. Cooling water is taken from the east slip through concrete tunnels and is discharged from the condenser into similar tunnels which empty into the west slip. (See Fig. 438.)

The dry vacuum pumps are of the rotative type, with cylinders 26 × 24, r.p.m. 100–120, and are driven by a 75-horse-power Corliss engine.

The circulating pumps are of the volute single-stage centrifugal type and are mounted on an extension of the main shaft of the engine driving the dry vacuum pump. They are rated at 22,500 gallons per minute each.

The hot-well pumps are of the two-stage centrifugal type, driven by 20-horse-power direct-current motors.

The feed pumps are of the Dean vertical single-cylinder pattern and are installed in duplicate for each unit. The steam cylinders are 24 inches in diameter, water cylinders 14 inches in diameter, and common stroke 24 inches. Feed water is drawn in by suction from the hot well at a temperature of about 100 degrees F. and is forced through closed heaters having 3000 sq. ft. of heating surface, and its temperature is raised to 180 degrees. The heater receives the steam exhausted from the steam-driven auxiliaries.

From the heater this water is forced through a 5-inch feed main to the different boilers in the section. The branches from main to boiler drum are 3 inches in diameter and fitted with an angle stop valve, a regrinding check and a grate-regulating valve. There is a 5-inch auxiliary main which supplies cold water to the boiler in case the main header is shut down.

Fig. 443. General Plan of Quarry Street Station, Units 1 and 2.

Miscellaneous. — The work is divided into eight-nour shifts. The list of operating men per unit is as follows:

In turbine room, including janitor work	2.0
In oil room	1.0
Attending water	0.5
Fireman	1
Fireman's helper	1
Conveyor men	2
Turbine switchboard gallery	0.3
Exciter tenders	0.2
Switchboard attendants	0.2
	8.2

Drains and drips from the auxiliaries empty into sumps from which they are discharged by Yeoman's bilge pumps into the discharge tunnel.

A steam-driven house pump is located in the basement.

The fire-protection system includes a 220-horse-power motor-driven spherical pump located in the basement and a connection for a fire tug.

Dining room, reading room, shower and tub baths and sleeping rooms for emergencies are provided for the employees.

Quarry Street Station. — Figs. 443 to 445 give general views of the Quarry Street Station, which is located directly across the river from the Fisk Street Station. The two stations are distinct; a breakdown in one would not affect the other; nevertheless, they are operated together. That is, there is one chief engineer for the two, and the combined station force of 350 men can be shifted from one to the other as needed.

The general layout of the Quarry Street Station differs from that of the Fisk Street Station on account of real-estate limitations. The boilers are in two parallel rows instead of the equipment for each unit extending at right angles to the turbine room as at Fisk.

The complete station will contain six 14,000-kilowatt Curtis turbo-alternators, the steam for each unit being supplied by eight 500 horse-power B. & W. boilers arranged as shown in Fig. 443. Steam is generated at 225 pounds gauge pressure and 150–175 degrees superheat. The settings for the first two units are similar to that illustrated in Fig. 67a, and the other similar to the one illustrated in Fig. 67b.

A novel system of ventilation enables the generators to be operated continuously at full load. As will be seen from Fig. 444 air ducts lead from an outside intake to the top of each unit, the revolving portion of the generator being designed to draw in a continuous supply of air and discharge it through openings in the turbine casing.

Everything is in duplicate, so that the chance of breakdown is remote There are two volute circulating pumps, each driven by a 125 horse-

FIG. 444. Side Elevation, Quarry Street Station, Units 1 and 2

power Corliss engine. The water of condensation is removed from the condensers by two rotary pumps driven by Kerr steam turbines. For each two units of turbines and boilers three horizontal boiler feed pumps are provided, located between the turbines as stated. There are also four step-bearing oil pumps, two oil accumulators, dry-air pumps, oil filters, etc. All are in plain view of the turbine-room operating force.

For the six ultimate units five 150-kilowatt exciters will be installed, three driven by horizontal Curtis steam turbines and two by 25-cycle, 220-volt induction motors. One of each of these types is included in the equipment of the present plant. In addition there is an excitation storage battery of 70 cells in the basement. Furthermore, in an emergency the split-pole rotary converter of the substation could be used for excitation.

Northwest Stations. — Two new stations of 120,000 kilowatt rated capacity each are to be installed on the north branch of the Chicago River near Roscoe Street and California Avenue. Each station is to be equipped with six 20,000-kilowatt Curtis turbo-generators, 2300 volt, 25 cycle, three phase, 750 r.p.m., similar to those installed at Quarry Street. The first two units for one station are now in course of erection. Each unit is to be supplied with steam at 250 pounds gauge pressure and 150 degrees superheat from ten 500-horse-power B. & W. boilers. The boiler settings are to be similar to the one illustrated in Fig. 67b. The general layout will be the same as at Fisk, the boiler lanes extending at right angles to the turbine room. There will be one chimney 17 feet inside diameter and 250 feet in height for every ten boilers.

The present capacity (July, 1910) of the Commonwealth Edison Company is about 240,000 kilowatts, divided as follows:

	Units.	Present Capacity.	Ultimate Capacity.
Fisk	10–12,000	120,000	120,000*
Quarry	6–14,000	84,000	84,000
Northwest No. 1	6–20,000	....	120,000
Northwest No. 2	6–20,000	....	120,000
Miscellaneous Plants	....	36,000	36,000
		240,000	480,000

* Space is available for four additional units, but no increase is contemplated at present.

COMPARATIVE BOILER ROOM AND ENGINE ROOM AREAS.

	Fisk.	Quarry.	Northwest.
Boiler room, sq. ft. per kw	0.51	....	0.44
Engine room, sq. ft. per kw	0.24	0.18	0.15
Total area, sq. ft. per kw	0.75	....	0.59

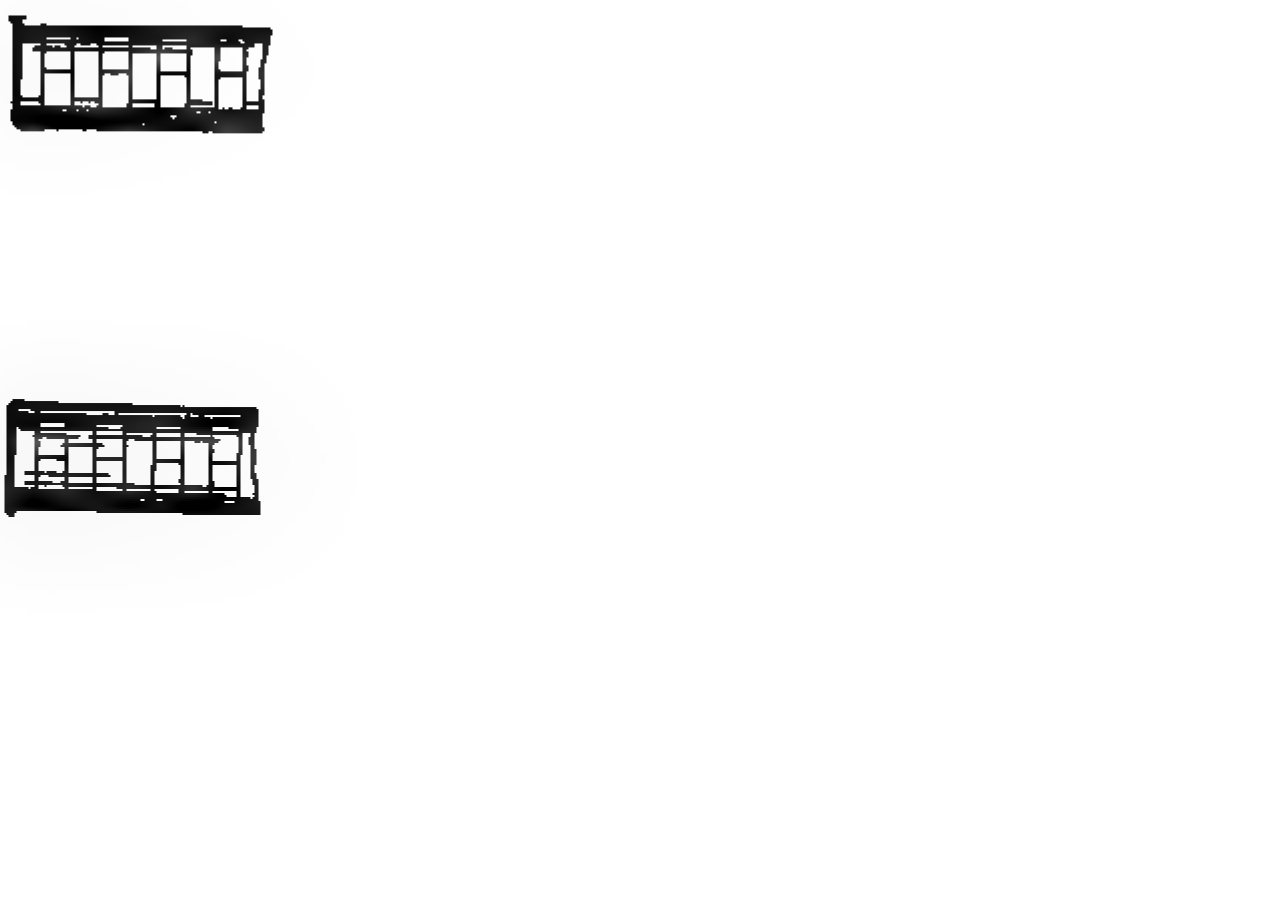

FIG. 445. Sectional Elevation, Quarry Street Station.

CHAPTER XXI.

A TYPICAL ISOLATED STATION. — MANUFACTURING PLANT.

The West Albany Power Station.*

THIS modern and well-designed station was erected (1904) in place of an old and practically worn-out plant, and is an example of what may be accomplished with a limited appropriation.

The power generated is used about the repair and construction shops of the New York Central Railroad Company at West Albany, N.Y., for lighting and general power purposes, and steam is used for heating. The equipment includes two direct-connected 600-kilowatt generators and two direct-acting air compressors.

General. — The plant (Fig. 446) is housed in a brick building 92 feet 8 inches wide by 113 feet 4 inches long, divided by a fire wall into an engine and a boiler room, the former being 46 feet wide.

The main units consist of two cross-compound non-condensing Ball & Wood engines and two Chicago Pneumatic Tool Company's air compressors. Steam is supplied by two batteries of horizontal water-tube boilers, comprising four boilers of 500 horse power each, made by the Franklin Boiler Works Company. These boilers have a guaranteed normal rating of 528 horse power under a natural draft not exceeding $\frac{7}{10}$ inch of water measured in the flue adjacent to the boiler, and an evaporation of not less than 9 pounds of water per pound of coal from and at 212 degrees F. They contain 5280 square feet of heating surface and 84 square feet of grate surface, a ratio of 63 to 1. There are 300 $3\frac{1}{2}$-inch tubes 13 feet long, surmounted by two 48-inch steam drums. The steam drums are connected by a 5-inch balance pipe on which two safety valves are mounted.

The boilers are suspended from wrought-iron and steel frames entirely independent of the brickwork, eliminating strains due to expansion and contraction.

The smoke flues run over the boilers to the stack placed between the two batteries, as shown in Figs. 447 and 448. Dampers are placed in both flues and hoods, the former operated by automatic regulators and the latter by hand, although the flue dampers may be readily disconnected and also operated by hand.

* See *Power*, November, 1904.

The steam nozzles are connected to dry pipes in each drum and to elbows outside, between which is a cross-over pipe carrying the main 10-inch valve, which is of the automatic, non-return, hand and emergency angle type. Between the angle stop valve and the main near the elbow is a gate valve. Long bends in the 10-inch pipe between stop valve and main are relied on for the necessary flexibility, and the main header is anchored to the stack near its middle point, as seen in Fig. 450, and its ends allowed free movement.

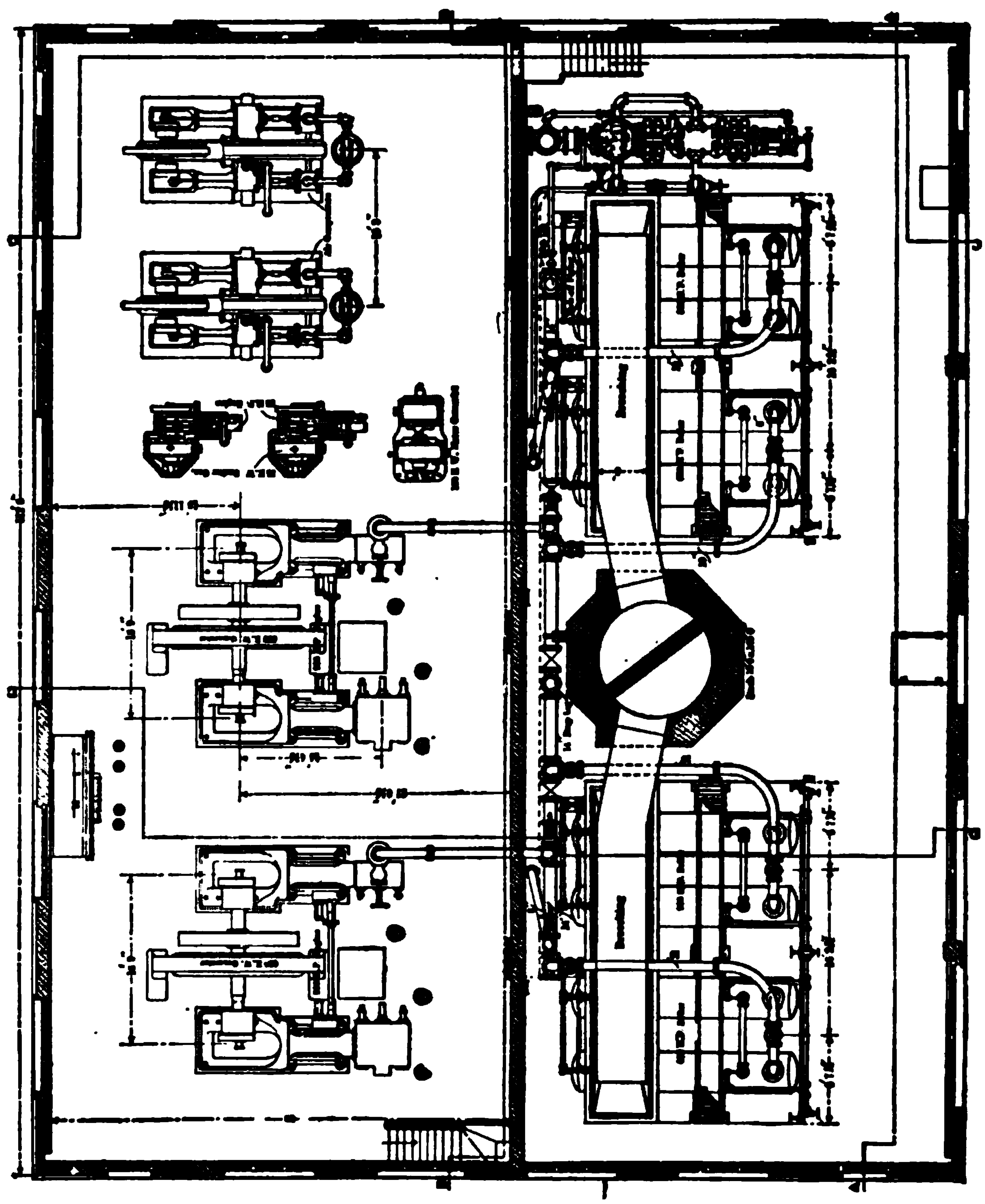

FIG. 446. Plan of Ground Floor.

A specially designed roller suspension is used on the larger pipes, and where pipes are swung from the floor beams a turn-buckle is provided in each suspension rod for adjustment.

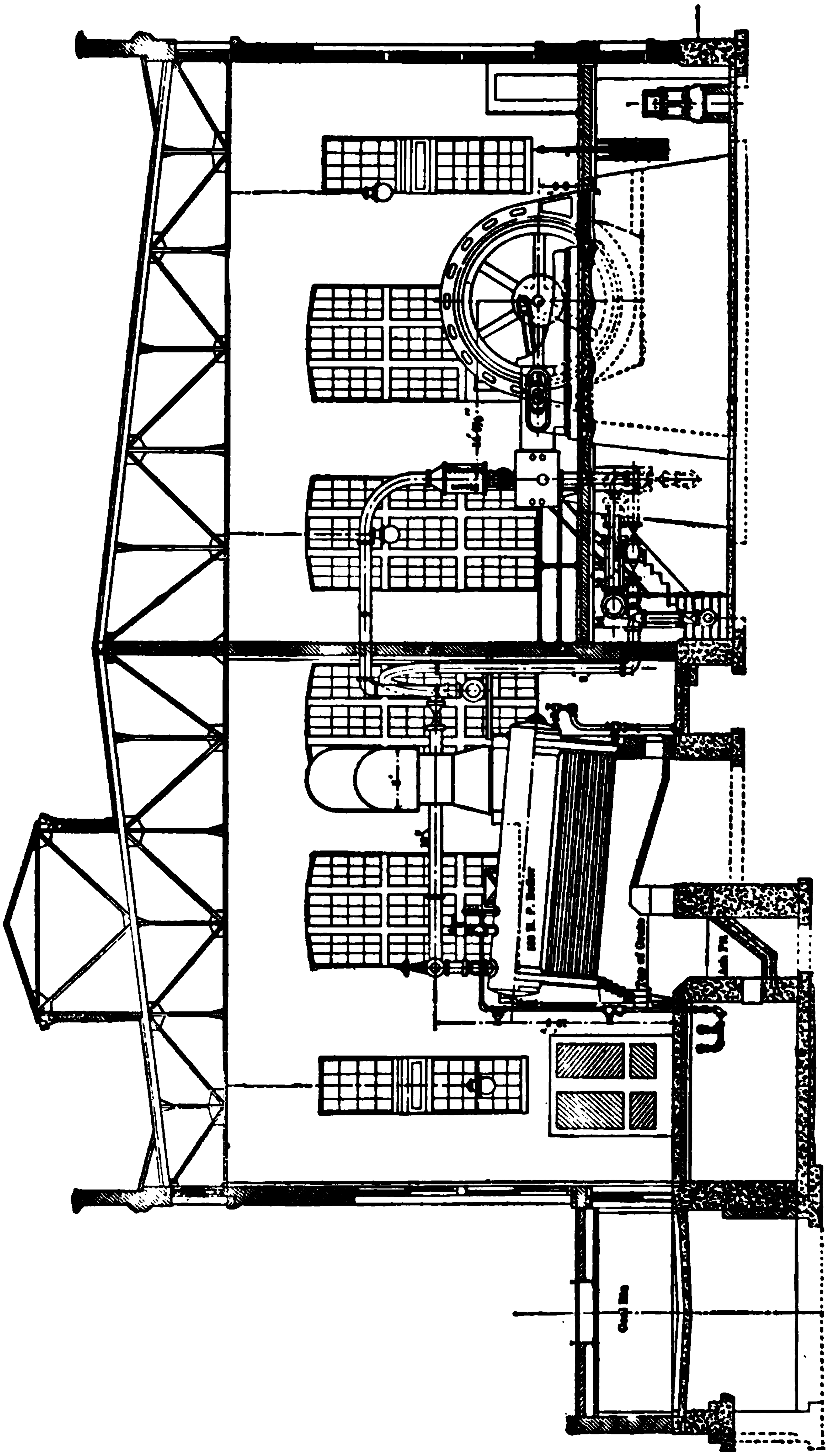

Fig. 447. Sectional Elevation through Line DD of Fig. 446.

FIG. 448. Sectional Elevation through Line CC of Fig. 446.

The steam piping slopes so as to drain to the drop legs, the drips being returned to the boiler by the Holly return system. Steam is led to the two large engines through 9-inch pipes, as shown in Figs. 449 and 450, a separator being placed just above each engine throttle.

Engines. — The engines have cylinders 21 and 41 inches in diameter and a stroke of 30 inches. They develop the normal load of 900 horse power, cutting off at 37 per cent of the stroke in each cylinder with steam at 175 pounds pressure and running at 120 revolutions per minute. The guaranteed steam consumption at this load is 19 pounds per indicated horse power. The generators, made by the General Electric Company, are mounted between the cylinders and supply three-phase alternating current, 60 cycles per second at 480 volts.

Speed regulation is sufficiently close to allow parallel operation of the generators, so that current is delivered to one bus-bar on the main switchboard in the engine room. Exhaust steam from the high-pressure cylinder is led to a vertical receiver below the engine-room floor, as shown in Figs. 449 and 450, and from there to the low-pressure cylinder. The low-pressure cylinders exhaust through 16-inch pipes into a 24-inch main extending nearly the whole length of the power house. This main also receives the exhausts from the exciter units, air compressors, and auxiliaries.

Exciters. — The two steam-driven exciter units consist of 35-kilowatt dynamos direct connected to 60-horse-power Woodbury engines made by the A. D. Granger Company. The engines are 7 x 12 single-cylinder automatic, and at 300 revolutions per minute and a normal cut-off of 20 per cent develop 58 horse power.

Besides these steam-driven units is a 100-kilowatt General Electric motor-generator. Fig. 451 gives a diagrammatic outline of the switchboard connections.

The two air compressors were made by the Chicago Pneumatic Tool Company and are of the cross-compound, two-stage type. The steam cylinders are 16 and 27 inches in diameter, the air cylinders are 14 and 24 inches in diameter, the common stroke 18 inches, and the capacity 1225 cubic feet of free air per minute.

Steam Piping. — Steam is supplied to the exciter engines and air compressors through an auxiliary steam line which runs nearly the length of the power house in the basement beneath the engine room as shown in Figs. 449 and 450. It is anchored near each end, and the long radius vertical bend near its middle point allows for expansion. It draws steam from the main steam pipe in the boiler room through the vertical pipe shown in Figs. 446 and 447. This pipe is taken out of

FIG. 449. Longitudinal Section through BB of Fig. 446.

the main at the top, makes a semicircular bend and passes through the wall between the engine and boiler rooms and enters the auxiliary main at the top.

Connections are made from the auxiliary main to the low-pressure cylinders of the 900-horse-power compound engines so that they may

FIG. 450. Plan of Basement.

be supplied with high-pressure steam should an occasion arise which would demand it.

The use of an auxiliary main may at first seem extravagant, but a careful study of the piping arrangement will show that it does not entail the use of more pipe or fittings and is a very desirable arrange-

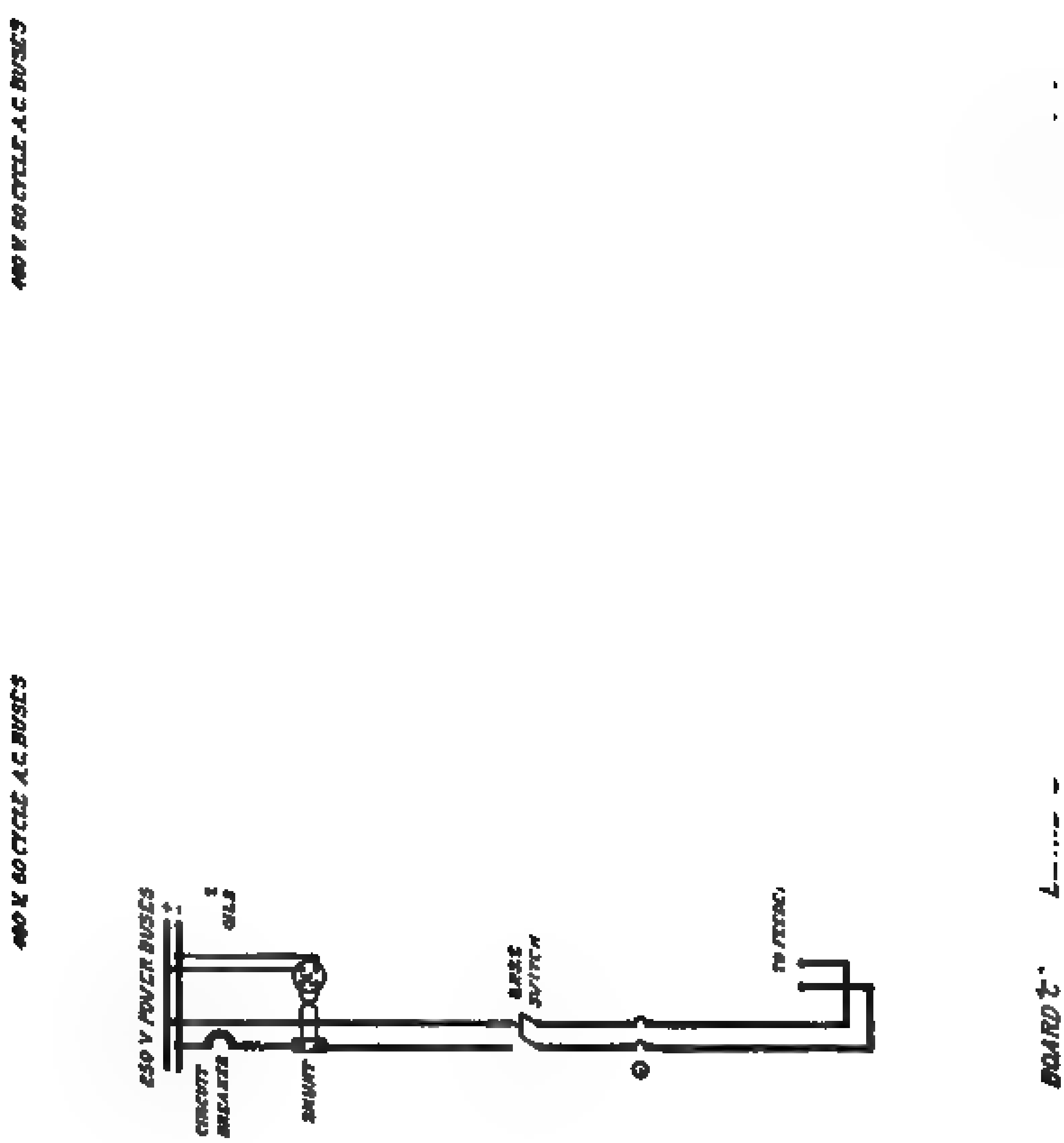

FIG. 451. Diagram of Switchboard Connections.

ment, particularly since it leaves the lines to the main engines intact and direct.

Connections to the other engines are direct, practically straight, and simple, and it is easy to make connections in both the steam and exhaust lines with any other apparatus which may be added in the future. In fact, the simplicity and yet completeness of the piping scheme is perhaps one feature of the plant.

Connection is made at the left of the auxiliary steam line, as shown in Figs. 449 and 450, to a steam supply for the shops at high pressure, or through a reducing valve at low pressure for heating.

The exhaust line, which runs nearly the length of the building in the basement and receives exhaust steam from all sources, shunts part of it through an exhaust muffler and oil separator to the shops for heating and the rest of it through an open heater and out of the exhaust pipe through the roof (Figs. 448 to 450). Either the muffler or the heater may be by-passed and the exhaust caused to flow directly to the heating system or the atmosphere.

Expansion in the exhaust line is provided for by two copper expansion joints, and an approved exhaust head is placed on the line extending above the roof.

All high-pressure steam and water piping is of "special full weight" lap-welded pipe and all low-pressure pipe of "standard" weight. Steam and hot-water pipes are covered with the best non-conducting covering. The high-pressure joints are made up with Merworth copper gaskets.

Two duplex Worthington feed pumps, made by the International Steam Pump Company, supply the boiler feed through the lines shown in Figs. 446, 447, and 450. The source of water supply is the city mains; hence the use of the meters.

Figs. 447 and 450 show the arrangement of the blow-off piping. There is a system of piping for fire protection which is connected to the city supply and runs in gradually reduced sizes as shown in Fig. 450. This system is laid out so as to drain itself through an arrangement of pipes discharging into the sewer.

Chimney. — A sectional view of the stack which furnishes draft for all of the boilers is shown in Fig. 446. It is 10 feet in diameter on the inside and 165 feet high, and designed to provide a draft of 50 per cent above the normal rating of the boilers, equivalent to 1.1 inches of water with the temperature of the heated gases not over 500 degrees F. It is made of radial bricks and has a baffle wall through the center for a short distance from the base to prevent the gases from the flues on each side from impinging and causing eddies. The wall is 20½ inches

thick at the base and 7½ at the top. The stack was built by M. W. Kellogg & Co.

The appropriation for the power plant did not allow the installation of a complete coal and ash-handling apparatus, but provision was made for such an installation in the future, and for the present the apparatus shown in Fig. 448 is used. The ashes fall from the grate to the ashpit (Fig. 447), and are raked out through the door into a barrow in the adjacent passageway, from which they are discharged into the vertical conveyor (Fig. 448) and transferred to a hopper and chute suspended from the roof beams and from there through a spout to cars outside. The coal is delivered over this track into the coal bin and carried to the boiler fronts in barrows. It is seen that the arrangement allows for the installation of a complete coal and ash-handling apparatus of the continuous type, and indeed one of the features of design is the provision for the installation of modern appliances in the future which are not possible at present.

The general arrangement of the boilers, engines, and piping is such that additional units of each may be installed by enlarging the building at one end without disarrangement of the apparatus already in place.

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Henry Heide Candy Company, *Engineering Record*	51: 269	May 20, 1905
Ingersoll-Sargent Drill Company, *American Electrician*	17: 237	May, 1905
Power	25: 323	June, 1905
Littleton Creamery, Denver, *Engineering Record*	50: 159	Aug. 6, 1904
Laray Mills, Gastonia, North Carolina, *Engineer* (United States)	40: 222	March 16, 1903
Ladew Factory, *Power*	25: 468	Aug., 1905
Patton Paint Company, Newark, New Jersey, *Electrical World*	42: 454	Sept. 12, 1903
Engineer (United States)	40: 798	Oct. 15, 1903

HYDRAULIC —*Continued.*

Modern Malt House, *Western Electrician*	37: 101	Aug. 5, 1905
John Mehl Factory, Jersey City, *Engineer* (United States)	38: 245	July 1, 1901
Manchester, New Hampshire, Mills, *Electrical World*	41: 269	Feb. 14, 1903
Monarch Mills, Union, South Carolina, *Engineering Record*	50: 380	Sept. 24, 1904
Lake Superior Company, St. Mary's Falls, Michigan, *Engineering News*	40: 68	Aug., 1898
Montmorency Falls, Canada, *American Electrician*	12: 553	Dec., 1900
Missouri River Power Company, *American Electrician*	14: 323	July, 1902
Montreal Power Plant, *Engineer* (London)	101: 130	Feb. 9, 1906
Monterey Gas and Electric Company, *Engineer* (United States)	42: 87	Jan. 16, 1905
Montgomery, Alabama, *Engineering News*	46: 418	Dec. 5, 1901
Massena, New York, Power Plant, *Engineering News*	45: 130	Feb. 21, 1901
Engineering Record	41: 2	Jan. 6, 1900
	122	Feb. 10, 1900
Western Electrician	22: 22	Jan. 8, 1898
Mexican Light and Power Company, *Engineering Record*	51: 575	May 20, 1905
Power Plant of the Milano-Gallarate-Porto Ceresio Railway, *Engineer* (United States)	40: 25	Jan. 1, 1903
American Electrician	14: 556	Dec., 1902
New Water Power Transmission Plants (C. L. Fitch), *Cassier's*	19: 243	Feb., 1901
North Mountain Power Company, *Engineer* (United States)	43: 123	Feb. 1, 1906
Engineering Record	53: 27	Jan. 6, 1906
New Milford, Connecticut, *Engineering Record*	49: 187	Feb. 13, 1904
	230	Feb. 20, 1904
Northern California Power Company, *Engineering Record*	50: 506	Oct. 29, 1904
Niagara (J. E. Woodbridge), *American Electrician*	12: 1	Jan., 1900
Transactions American Society of Mechanical Engineers	19: 839	June, 1898
Extension of Niagara Falls Power Plant, *American Machinist*	21: 463	June 23, 1898
Western Electrician	22: 137	March 5, 1898
Niagara Falls Power (H. W. Buck), *Cassier's*	20: 1	May, 1901
	179	Jan., 1902
	25: 103	Dec., 1903
Engineering	74: 637	Nov. 12, 1902
Engineering News	48: 9	July 3, 1902
	250	Oct. 2, 1902
	490	Dec. 11, 1902

HYDRAULIC — *Continued.*

Power Plant of the Niagara Falls (Coleman Sellers), *Engineering*	67:91	Jan. 20, 1899
	128	Jan. 27, 1899
	160	Feb. 3, 1899
Canadian District Plant (C. B. Smith), *Engineering Magazine*	28:727	Feb., 1905
New Wheel Pit Niagara Falls Power Company (O. E. Dunlap), *Engineering News*	43:229	April 5, 1900
Engineering Record	43:150	Feb. 16, 1901
Power of Lower Niagara (O. E. Dunlap), *Western Electrician*	32:360	June 18, 1898
Niagara Falls (L. B. Stillwell), *Western Electrician*	29:191	Sept. 21, 1900
Progress on Power House at Niagara (O. E. Dunlap), *Western Electrician*	29:234	Oct. 12, 1901
Niagara Power Plant of Ontario, *Engineering News*	54:475	Nov. 9, 1905
	561	Nov. 30, 1905
Power Station at Neuchatel, *American Electrician*	16:493	Oct., 1904
Norwegian Hydro-Electric Plant, *Electrical World*	46:135	July 22, 1905
Western Electrician	37:63	July 22, 1905
Hydro-Electric Power Plant of Christiania, Norway, *Engineer* (United States)	40:227	March 16, 1903
Works of the Ontario Power Company, *Engineering Record*	50:420	Oct. 8, 1904
	460	Oct. 15, 1904
	480	Oct. 22, 1904
	29	Oct. 29, 1904
Oliver Plow Company, *Engineer* (United States)	42:363	June 1, 1905
Ontario Power Company, *Engineering Record*	50:420	Oct. 8, 1904
	460	Oct. 15, 1904
	480	Oct. 22, 1904
	504	Oct. 29, 1904
Ottawa-Ontario Power Company, *Western Electrician*	27:85	Aug. 11, 1900
Port Huron Light and Power Company, *Engineering Record*	49:458	April 9, 1904
The Puyallup River Water Power Development, *Engineering News*	52:273	Sept. 29, 1904
Engineering Record	50:399	Oct. 1, 1904
Saut Mortier Transmission Plant, *American Electrician*	17:457	Sept., 1905
The Sill Hydraulic Power Plant near Innsbruck, *Engineering Record*	52:13	July 1, 1905
Standard and Bay Company Plant, *Engineer* (United States)	40:46	Jan. 1, 1903
Snoqualmie Falls, *American Electrician*	13:497	Oct., 1901
Power	19:1	Oct., 1899
Engineering	44:398	Dec. 13, 1900

HYDRAULIC — *Continued.*

Swiss Combined Water Power and Gas Engine Plant, *American Electrician*	15:113	March, 1903
St. Croix Power Plant, *Engineering Magazine*	20:954	Feb., 1901
Sault Sainte Marie Power Plant (H. Von Schon), *Engineering Magazine*	24:273	Nov., 1902
Combined Hydraulic and Steam Plant, Stuyvesant Falls, New York, *Engineering Record*	43:3	Jan. 5, 1901
Sterling Hydraulic Company, *Engineering Record*	52:688	Dec. 16, 1905
St. Maurice-Lausanne, Switzerland, *Western Electrician*	33:477	Dec. 20, 1903
Sewanee Falls Power Plant, *Engineering Record*	53:44	Jan. 13, 1906
Shawinigan Falls, Canada, *Cassier's*	26:202	July, 1904
Engineering Record	41:391	April 29, 1900
Hydro-Electric Development at Turner's Falls, *Electrical World*	46:263	Aug. 12, 1905

STEAM ENGINE PLANTS.

Aurora, Elgin and Chicago Railway, *Street Railway Journal*	19:143	Feb., 1902
	20:574	Oct. 4, 1902
Augusta Railway Power Station, *Street Railway Journal*	21:24	Jan. 3, 1903
Brooklyn Rapid Transit Company, *Electrical World*	46:519	Sept. 23, 1905
Engineering Record	52:254	Sept. 23, 1905
Street Railway Journal	21:256	Feb. 14, 1903
	26:432	Sept. 23, 1905
Street Railway Review	14:332	May 20, 1904
Berlin Elevated and Underground Railway, *Engineering Record*	45:389	April 26, 1902
Boston Elevated Power Station, *Street Railway Journal*	17:253	March, 1901
	20:118	July 26, 1902
Boston and Manhattan Railway, *Street Railway Journal*	20:554	Oct. 4, 1902
Some British Central Electric Power Stations (H. F. Parshall), *Cassier's*	26:233	June, 1904
Breslau, Germany, *Power*	24:742	Dec., 1904
Brigham City, Utah, Light Plant, *Engineering News*	44:235	Sept. 7, 1905
Boston Elevated, Lincoln Wharf Station, *Engineer* (United States)	40:4	Jan. 1, 1903
Berlin Electric Railway, *Engineer* (United States)	40:16	Jan. 1, 1903
Brussels Electric Power Station at Anderlecht, *Engineer* (United States)	41:555	Aug. 15, 1904
		Oct. 14, 1905
Boston and Worcester Street Railway, *Engineer* (United States)	40:507	July 1, 1903
Engineering Record	52:438	

Steam Engine Plants—*Continued.*

Brooklyn Navy Yard, *Power*	20:21	Dec., 1900
Berkshire Street Railway Company, *Street Railway Review*	12:813	Nov. 20, 1902
American Electrician	15:387	Aug., 1903
Engineering Record	46:74	July 26, 1902
Brussels Electric Tramways, *Power*	25:1	Jan., 1905
Bloomington and Normal Railway, *Street Railway Journal*	25:934	May 27, 1905
Cranford, New Jersey, *Engineering Record*	49:355	March 19, 1904
Consolidated Traction Company, Pittsburg, *Street Railway Journal*	15:127	March, 1899
Street Railway Review	9:135	Feb. 15, 1899
Cleveland Electric Railway Company, *Street Railway Journal*	15:199	April, 1899
	267	May, 1899
	17:655	June, 1901
	19:500	April, 1902
	23:162	Jan. 30, 1904
Conneaut and Erie, *Street Railway Journal*	23:195	Feb. 6, 1904
Canton and Akron, *Street Railway Journal*	23:805	May 28, 1904
Cleveland, Painesville, and Ashtabula, *Street Railway Journal*	24:93	July 16, 1904
Chicago and Western Indiana, *Western Electrician*	37:425	Dec. 2, 1905
Cincinnati, Georgetown and Portsmouth Railway, *Engineer* (United States)	40:353	May 15, 1903
Clyde Valley Electric Power Company, *Engineer* (United States)	42:524	Aug. 1, 1905
Capitol Traction Company, Washington, District of Columbia, *Power*	19:1	Feb., 1899
Engineering Record	39:99	Dec. 31, 1898
Street Railway Journal	15:9	Jan., 1899
Citizens' Light and Power Company, *Power*	22:1	Feb., 1902
Chicago Electric Traction Company, *Street Railway Review*	8:184	March 15, 1898
Engineering News	39:2	Jan. 6, 1898
Chattanooga Electric Company, *Street Railway Review*	15:136	March 15, 1905
Carville (England) Power Plant, *Street Railway Journal*	21:902	June 20, 1903
Dublin United Tramways, *Power*	19:1	Jan., 1899
Detroit, Ypsilanti and Ann Arbor Railway, *Street Railway Review*	10:5	Jan. 15, 1900
Des Moines, Iowa, *Engineering Record*	46:25	July 12, 1902
Street Railway Journal	22:54	July 11, 1903
Denver Tramway System, *Street Railway Journal*	22:683	Oct. 10, 1903
Dayton and Muncie Electric Railway, *Street Railway Journal*	26:979	Dec. 2, 1905
Detroit United Railway, *Street Railway Journal*	20:445	Oct. 4, 1902

STEAM ENGINE PLANTS — *Continued.*

Everett Railway and Electric Company, *Engineer* (United States)	40: 839	Nov. 16, 1903
Exeter, Hampton and Amesbury Street Railway, *Street Railway Review*	10: 630	Nov. 15, 1900
Edison Electric Company, Atlantic Avenue Station, Boston, *Transactions American Society of Mechanical Engineers*	23: 569	May, 1902
Glasgow Municipal Tramways, *Power*	22: 1	Aug., 1902
Street Railway Journal	15: 247	April, 1899
	17: 625	June, 1901
Georgetown, Colorado, *Western Electrician*	27: 282	Nov. 3, 1900
Grand Rapids, Holland and Lake Michigan Railway, *Engineer* (United States)	39: 179	March 15, 1902
Hanover (Germany) Power Station, *Power*	25: 583	Oct., 1905
Western Electrician	38: 55	Jan. 20, 1906
Hartford and Springfield Railway, *Engineering Record*	45: 485	May 24, 1902
Halifax, England, Tramway, *Street Railway Journal*	20: 293	Sept. 6, 1902
Harrisburg, Pennsylvania, *Street Railway Journal*	27: 61	Jan. 6, 1906
Hazleton, Pennsylvania, *Street Railway Journal*	21: 350	March 7, 1903
Hebron, Ohio, *Street Railway Journal*	22: 147	Aug. 1, 1903
Interborough Rapid Transit Company, *American Electrician*	16: 501	Oct., 1904
Engineering Record	51: 345	March 25, 1905
	50: 384	Oct. 1, 1904
	424	Oct. 8, 1904
	456	Oct. 15, 1904
	490	Oct. 22, 1904
	510	Oct. 29, 1904
	541	Nov. 5, 1904
Western Electrician	35: 211	Sept. 17, 1904
Power	24: 511	Sept., 1904
Indianapolis and Cincinnati Traction Company, *Engineering Record*	51: 329	March 18, 1905
Street Railway Journal	25: 502	March 18, 1905
Interurban Railway and Terminal Company, Cincinnati, *Engineer* (United States)	41: 251	April 1, 1904
Independent Electric Light and Power Company, San Francisco, *Power*	22: 1	March, 1902
Johnstown, Pennsylvania, Power Station, *Street Railway Review*	11: 426	July 15, 1901
Johnston and Gloversville Railway, *Street Railway Review*	22: 308	Aug. 22, 1903
Kankakee Power and Light Plant, *Western Electrician*	26: 127	March 3, 1900
Kingsbridge Power Station, *Engineering Record*	50: 10	July 2, 1904
Engineering News	43: 189	March 22, 1900
Central Station at Kien, *Engineer* (United States)	42: 337	May 15, 1905

STEAM ENGINE PLANTS—*Continued.*

Kaw River Power Station, Kansas City, *Street Railway Review*	10:560	Oct. 15, 1900
Louisville and Nashville Railway Company, South Louisville, Kentucky, *Engineer* (United States)	42:499	Aug. 1, 1903
Louisville Railway Company, *Engineer* (United States)	42:511	Aug. 1, 1905
Street Railway Review	9:439	July 15, 1899
LaBella Power Plant, Goldfield, Colorado, *Western Electrician*	25:267	Nov. 4, 1899
Lisbon Railway Power Station, *Street Railway Journal*	17:293	March, 1901
Liverpool Electric Railway, *Street Railway Journal*	17:390	April, 1901
London Power Stations, *Street Railway Journal*	18:287	Sept., 1901
Lackawanna and Wyoming Valley Railroad, *Engineering Record*	47:624	June 13, 1903
Street Railway Journal	21:868	June 13, 1903
London County Council Tramway, *Street Railway Journal*	21:821	June 6, 1903
Leicester Power Station, *Street Railway Journal*	23:832	June 4, 1904
Manchester Traction Company (W. V. Batson), *Engineering Record*	47:107	Jan. 24, 1903
Street Railway Journal	20:300	Sept. 6, 1902
	25:817	May 6, 1905
Muskegon, Michigan, Light and Traction Company, *Engineering Record*	48:452	Oct. 17, 1903
Metropolitan West Side Elevated, Chicago, *Street Railway Journal*	15:581	Sept., 1899
	709	
McKeesport and Connellsville Railway, *Engineering Record*	48:264	Sept. 5, 1903
Manx Railway, Isle of Man, *Street Railway Journal*	23:357	March 5, 1904
Milwaukee Gas Light Company, *Engineering News*	55:28	Jan. 11, 1906
Moulineaux Power Station, Paris, *American Electrician*	16:325	July, 1904
Power Station of the Metropolitan Railway, *Engineer* (United States)	37:271	Nov. 15, 1900
American Electrician	12:112	March, 1900
Monterey Gas and Electric Company, *Engineer* (United States)	42:87	Jan. 16, 1905
Market Street Railway Company of San Francisco, *Power*	19:3	March, 1899
Manhattan Elevated Railway, *Power*	21:1	April, 1901
Street Railway Review	9:82	Feb. 16, 1899
Engineering (United States)	39:80	Feb. 1, 1902
Engineering News	47:82	Jan. 30, 1902
Milford, Attleboro and Woonsocket Railway, *Street Railway Review*	10:638	

STEAM ENGINE PLANTS — *Continued.*

New York Central Power House, *Electrical World*...	46:95	July, 1905
Power..................................	24:228	April, 1904
Newcastle and District Electric Lighting Company, *Engineer* (United States).....................	40:29	Jan. 1, 1903
Northwestern Elevated Railroad, *Street Railway Review*.................................	15:199	
	267	
	17:655	
	19:500	
	23:162	
Newcastle, England, *Engineering Record*.........	46:157	Aug. 16, 1902
New York Gas and Electric Light and Power Company, *Engineering News*................	45:375	May 23, 1901
Oakland, California, *Engineering Record*..........	49:591	May 7, 1904
Omaha and Council Bluffs Railway and Bridge Company, *Street Railway Review*..............	9:296	May 15, 1899
Paris Metropolitan, *American Electrician*.........	16:111	March, 1904
Engineer (United States)..................	40:13	Jan. 1, 1903
Pan-American Exposition, *Engineer* (United States)..................................	38:158	June 1, 1901
Paris Exposition Plant, *Power*..................	19:7	Dec., 1899
Pittsburg Traction Company, *Street Railway Review*.................................	9:135	Feb. 15, 1899
	15:401	July 15, 1905
Philadelphia Rapid Transit Company, *Street Railway Review*............................	15:526	Sept. 15, 1905
Engineering Record........................	47:611	June 20, 1903
	52:340	Sept. 23, 1905
Electrical Power Development at Portland, Oregon, *Electrical World*....................	46:174	July 29, 1905
Engineering Record.......................	52:142	Aug. 5, 1905
Peekskill (New York) Light and Railway Company, *Street Railway Journal*......................	20:92	July 19, 1902
Pacific Electric Railway Company, Los Angeles, *Street Railway Journal*......................	23:394	March 12, 1904
Richmond, Virginia, *Engineering Record*..........	49:11	Jan. 2, 1904
Rock Island Railway Shops at East Moline, Illinois, *Engineering Record*..................	50:137	July 30, 1904
Rochester Power Plant, *Engineer* (United States).	38:312	Sept. 1, 1901
Scioto Valley Traction Company, *Engineering Record*.................................	50:644	Dec. 3, 1904
St. Louis Exposition Power Plant, *Engineering News*..................................	42:320	
American Electrician......................	16:528	Oct., 1904
Engineering News........................	42:223	Sept. 15, 1904
Western Electrician......................	35:303	Oct. 15, 1904
Power.................................	23:481	Nov., 1903
Saginaw and Bay City Light Company, *Engineer* (United States)...........................	42:399	June 15, 1905

STEAM ENGINE PLANTS—*Continued.*

St. Louis Transit Company, *Street Railway Journal*	11:813	Nov. 15, 1901
Engineering News	47:269	April 3, 1902
	297	April 10, 1902
Stark Electric Railway Company, *Street Railway Journal*	25:10	Jan. 7, 1905
The Sunderland District Tramways, *Street Railway Journal*	26:96	July 15, 1905
St. Clair Tunnel, Grand Trunk Railway, *Engineering News*	55:62	Jan. 18, 1906
Dayton, Springfield and Urbana Railway, *American Electrician*	12:415	Sept., 1900
Southern Pacific Company, Galveston, Texas, *American Electrician*	15:476	Oct., 1903
Syracuse Power Station, *Street Railway Journal*	19:517	May, 1902
Sydney, New South Wales, *Street Railway Journal*	20:930	Dec. 6, 1902
Seattle Power Station, *Street Railway Journal*	21:649	May 2, 1903
South Side Elevated, *Western Electrician*	26:231	April 14, 1900
Springfield, Illinois, Light and Power Company, *Electrical World*	47:253	Feb. 3, 1906
Toledo and Monroe, *Street Railway Journal*	18:124	Aug., 1901
Tokio Electric Railway, *Street Railway Journal*	19:245	March, 1902
Toledo, Port Clinton and Lakeside Railway, *Engineer* (United States)	43:295	March, 1906
Toledo and Western Railway, *Street Railway Journal*	20:980	Dec. 20, 1902
Twin City Rapid Transit Company, *Engineering Record*	50:692	Dec. 10, 1904
Street Railway Review	14:441	July 20, 1904
Terminal Railroad Association of St. Louis, *Engineering Record*	51:92	Jan. 28, 1905
Trans-Mississippi Exposition, Omaha, *Power*	18:7	June, 1898
Toledo and Maumee Valley Railway, *Street Railway Review*	9:613	Sept. 15, 1899
American Electrician	12:464	Oct., 1900
United Railways and Electric Company, Baltimore, *Engineer* (United States)	39:357	May 15, 1902
Union Traction Company, *Street Railway Review*	9:12	Jan. 15, 1899
Union Traction Company, Anderson, Indiana, *Engineering Record*	43:495	May 25, 1901
Street Railway Journal	18:826	Dec., 1901
United Electric Company, Hoboken, New Jersey, *Engineering Record*	46:56	July 19, 1902
Union Loop Power House, Chicago, *Western Electrician*	24:1	Jan. 7, 1899
Virginia Electric Railway and Equipment Company, *Engineering News*	45:318	May 2, 1901
Voltellina Three-Phase Railway, *Street Railway Journal*	21:791	May 30, 1903

STEAM ENGINE PLANTS — *Continued.*

Worcester and Blackstone Valley Street Railway, *Engineering Record*	47: 462	May 2, 1903
Warren and Jamestown Single-Phase Railway, *Street Railway Journal*	27: 270	Feb. 16, 1906
Electrical World	47: 363	Feb. 17, 1906
Washtenaw Electric Company, *Engineer* (United States)	38: 150	May 15, 1901
Willimantic Gas and Electric Company, *Engineer* (United States)	42: 163	March 1, 1905
West Albany Power Station, *Power*	24: 677	Nov., 1904
Wilkesbarre Interurban Railway, *Street Railway Review*	9: 131	Feb. 15, 1899
Weehauken Station New York Central and Hudson River Railroad, *American Electrician*	17: 501	Oct., 1905
Engineering News	54: 506	Nov. 16, 1905
Street Railway Journal	26: 872	Nov. 11, 1905
Engineering Record	52: 553	Nov. 11, 1905
Western Pennsylvania Railway and Light System, *Street Railway Journal*	20: 139	Aug. 2, 1902
Zanesville Railway and Light Company, *Electrical World*	43: 500	March 19, 1904

TURBINE PLANTS.

Amsterdam-Haarlem Electric Railway, *Engineering Record*	51: 18	Jan. 17, 1905
Street Railway Journal	25: 21	Jan. 7, 1905
Brooklyn Rapid Transit, Williamsburg Plant, *Street Railway Journal*	26: 432	Sept. 23, 1905
Generating Station Champ, *American Electrician*	17: 65	Feb., 1905
Clyde Valley Electric Power Company, *Engineering Record*	52: 209	Sept. 9, 1905
Chelsea Generation Station of the London Underground, *Engineering Record*	52: 215	Feb. 25, 1905
Power	23: 421	Aug., 1903
Street Railway Journal	25: 388	March 4, 1905
Commonwealth Electric Company, Fisk Street Station, *Western Electrician*	38: 55	Jan. 20, 1906
Power	26: 715	Dec., 1906
Dubuque, Iowa, Power Plant, *Engineering Record*	20: 202	Aug. 13, 1904
Detroit Edison Company, *Engineering Record*	52: 194	Oct. 7, 1905
DeBeers Consolidated Mines, *Engineering Record*	51: 4	Jan. 7, 1905
Edison Electric Illuminating Company, Boston, *Engineering Record*	51: 150	Feb. 11, 1905
Power	25: 389	July, 1905
Hartford Electric Light Company, Dutch Point Plant, *Engineering Record*	51: 204	Feb. 25, 1905
Long Island Railroad Power House, *Street Railway Journal*	25: 24	Jan. 7, 1905
Engineering Record	49: 454	April 9, 1904

Turbine Plants — *Continued.*

Los Angeles Edison Company, *Power*	26: 67	Feb., 1906
Mexican Central Shops at Aguascalientes, *Engineering Record*	50: 227	Aug. 20, 1904
	247	Aug. 27, 1904
Municipal Turbine Plant, Anderson, Indiana, *Engineer* (United States)	42: 641	Oct. 2, 1905
Manchester, England, Power Station, *Street Railway Journal*	25: 934	May 27, 1905
New York Edison Waterside Station, *Electrical World*	46: 383	Sept. 2, 1905
	435	Sept. 9, 1905
Power	22: 1	Jan., 1902
New York Central Steam-Electric Station, *Power*	26: 131	March, 1906
Engineer (United States)	43: 733	Nov. 15, 1906
New York and Long Island River Power Station, *Power*	26: 199	April, 1906
New Orleans Power House, *Power*	24: 651	Nov., 1904
Street Railway Review	9: 393	June 15, 1899
New Bedford Power Station, *Street Railway Review*	11: 884	Dec., 1901
Old Colony Street Railway, Quincy Point Station, *Engineering Record*	51: 646	June 10, 1905
Street Railway Journal	25: 1022	June 10, 1905
Engineer (United States)	43: 85	Jan. 15, 1906
Steam Turbine Plant at Poughkeepsie, New York, Shop, *Engineering Record*	51: 454	April, 1905
Potomac Electric Power Company, Washington, District of Columbia, *Power*	27: 277	May, 1907

Isolated Stations.

Apartments.

Collingwood Apartment Hotel, New York, *Engineering Record*	45: 323	April 5, 1902
Ansonia Apartment, New York, *Engineering Record*	46: 467	Nov. 15, 1902

Hospitals.

Agnes Memorial Sanatorium, *Engineering Record*	50: 312	Sept. 10, 1904
Connecticut Hospital for the Insane, *Engineering Record*	52: 44	July 8, 1905
Lakeside Hospital, Cleveland, Ohio, *Engineer* (United States)	39: 108	Feb. 15, 1902
Massachusetts General Hospital, Boston, *Transactions American Society of Mechanical Engineers*	22: 392	Jan., 1901

Hotels.

Bellevue-Stratford Hotel, Philadelphia, *Engineering Record*	51: 14	Jan. 7, 1905

HOTELS — *Continued.*

Hotel Belmont, New York, *Engineering Record*	52: 739	Dec. 30, 1905
	53: 9	Jan. 6, 1906
	56	Jan. 13, 1906
	81	Jan. 20, 1906
Hotel Gotham, New York, *American Electrician*	17: 551	Nov., 1905
Engineering Record	52: 517	Nov. 4, 1905
New York Athletic Club, *Electrical World*	31: 463	April 16, 1898
	585	May 14, 1898
University Club, *Engineering Record*	46: 36	July 12, 1902

MANUFACTURING PLANTS.

American Steam Pump Company, *Engineer* (United States)	39: 217	April 1, 1902
Atlas Knitting Mills, *Power*	25: 408	July, 1905
American Lithographic Company, *American Electrician*	10: 1	
Armour Packing Company (J. E. Smith), *American Electrician*	12: 202	May, 1900
Western Electrician	23: 305	Nov. 26, 1898
Anheuser-Busch Brewing Association, *Engineer* (United States)	42: 67	Feb. 1, 1905
Power	25: 84	Feb., 1905
Booth Cold Storage, *Engineering Record*	45: 97	Feb. 1, 1902
Columbian Cordage Company, *Engineering Record*	50: 447	Oct. 15, 1904
Power Plants for Cotton Mills, *Engineer* (United States)	41: 135	Feb. 15, 1904
Cascade Water Power and Light Company, Cascade, Canada, *Engineer* (United States)	40: 603	Aug. 1, 1903
Camden Interstate Railway Company, *Street Railway Review*	15: 269	May 15, 1905
DeCew Falls Power Plant, *Western Electrician*	38: 115	Feb. 10, 1906
DeSabla, California, Station, *Engineering News*	54: 131	Aug. 10, 1905
Dan River Power and Manufacturing Company's Plant, *Engineering Record*	50: 291	Sept. 3, 1904
Power Plants of Edison Electric Company of Los Angeles, *Engineering Record*	51: 211	Feb. 25, 1905
	302	March 11, 1905
	325	March 18, 1905
Electric Generating Station at Glomen, near Kykkelsrud, *American Electrician*	17: 409	Aug., 1905
Engineering Record	50: 19	July 2, 1904
Elgin Watch Works, Elgin, Illinois, *Engineering Record*	56: 294	Sept. 14, 1907
Hydro-Electric Power Development for Guanajuato, Mexico, *Engineering Record*	50: 195	Aug. 13, 1904
General Electric Company's Hudson River Plant, *Electrical World*	43: 1115	June 11, 1904
Garvins Falls, New Hampshire, *Engineering Record*	49: 668	May 28, 1904

MANUFACTURING PLANTS — *Continued.*

Grésivaudan Valley Power Plant, *Western Electrician*		
Hudson River Company, Mechanicsville, New York	23: 168	Oct. 1, 1898
Engineering News	40: 130	Sept. 1, 1898
Western Electrician	23: 135	Sept. 3, 1898
Hampton, Virginia, *Engineering Record*	48: 179	Aug. 15, 1903
Indian Power Plant, *Engineer* (London)	101: 36	Jan. 12, 1906
Engineering	81: 103	Jan. 19, 1906
Jhelum Power Plant, *Engineer* (United States)	43: 240	March 15, 1906
Kern River Company's Enterprise, *Engineering News*	52: 55	July 21, 1904
Little Falls, Montana, Power Station, *Engineering Record*	51: 616	June 3, 1905
McCormick Twine Mills, *Western Electrician*	28: 109	Feb. 16, 1901
National Cash Register Company, Dayton, *Engineer* (United States)	39: 136	March 1, 1902
New England Confectionery Company, *Engineer* (United States)	41: 231	April 1, 1904
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APPENDIX B.

RULES FOR CONDUCTING BOILER TRIALS.*

Code of 1899.

I. *Determine at the outset* the specific object of the proposed trial, whether it be to ascertain the capacity of the boiler, its efficiency as a steam generator, its efficiency and its defects under usual working conditions, the economy of some particular kind of fuel, or the effect of changes of design, proportion, or operation; and prepare for the trial accordingly. (Appendix II.)

II. *Examine the boiler*, both outside and inside; ascertain the dimensions of grates, heating surfaces, and all important parts; and make a full record, describing the same, and illustrating special features by sketches. The area of heating surface is to be computed from the surfaces of shells, tubes, furnaces, and fire boxes in contact with the fire or hot gases. The outside diameter of water tubes and the inside diameter of fire tubes are to be used in the computation. All surfaces below the mean water level which have water on one side and products of combustion on the other are to be considered as water-heating surface, and all surfaces above the mean water level which have steam on one side and products of combustion on the other are to be considered as superheating surface.

III. *Notice the general condition* of the boiler and its equipment, and record such facts in relation thereto as bear upon the objects in view.

If the object of the trial is to ascertain the maximum economy or capacity of the boiler as a steam generator, the boiler and all its appurtenances should be put in first-class condition. Clean the heating surface inside and outside, remove clinkers from the grates and from the sides of the furnace. Remove all dust, soot, and ashes from the chambers, smoke connections, and flues. Close air leaks in the masonry and poorly fitted cleaning doors. See that the damper will open wide and close tight. Test for air leaks by firing a few shovels of smoky fuel and

* From the report of the Committee of the American Society of Mechanical Engineers on the revision of the Society Code of 1885 relative to a standard method of conducting steam boiler trials.

immediately closing the damper, observing the escape of smoke through the crevices, or by passing the flame of a candle over cracks in the brickwork.

IV. *Determine the character of the coal* to be used. For tests of the efficiency or capacity of the boiler for comparison with other boilers the coal should, if possible, be of some kind which is commercially regarded as a standard. For New England and that portion of the country east of the Allegheny Mountains, good anthracite egg coal, containing not over 10 per cent of ash, and semi-bituminous Clearfield (Pa.), Cumberland (Md.), and Pocahontas (Va.) coals are thus regarded. West of the Allegheny Mountains, Pocahontas (Va.) and New River (W. Va.) semi-bituminous, and Youghiogheny or Pittsburg bituminous coals are recognized as standards.* There is no special grade of coal mined in the Western States which is widely recognized as of superior quality or considered as a standard coal for boiler testing. Big Muddy lump, an Illinois coal mined in Jackson County, Ill., is suggested as being of sufficiently high grade to answer the requirements in districts where it is more conveniently obtainable than the other coals mentioned above.

For tests made to determine the performance of a boiler with a particular kind of coal, such as may be specified in a contract for the sale of a boiler, the coal used should not be higher in ash and in moisture than that specified, since increase in ash and moisture above a stated amount is apt to cause a falling off of both capacity and economy in greater proportion than the proportion of such increase.

V. *Establish the correctness of all apparatus* used in the test for weighing and measuring. These are:

1. Scales for weighing coal, ashes, and water.

2. Tanks, or water meters for measuring water. Water meters, as a rule, should only be used as a check on other measurements. For accurate work, the water should be weighed or measured in a tank. (Appendices I, IV, VII, VIII.)

3. Thermometers and pyrometers for taking temperatures of air, steam, feed water, waste gases, etc. (Appendix XXVII.)

4. Pressure gauges, draught gauges, etc. (Appendices XXVIII to XXX.)

* These coals are selected because they are about the only coals which contain the essentials of excellence of quality, adaptability to various kinds of furnaces, grates, boilers, and methods of firing, and wide distribution and general accessibility in the markets.

The kind and location of the various pieces of testing apparatus must be left to the judgment of the person conducting the test; always keeping in mind the main object, i.e., to obtain authentic data.

VI. *See that the boiler is thoroughly heated* before the trial to its usual working temperature. If the boiler is new and of a form provided with a brick setting, it should be in regular use at least a week before the trial, so as to dry and heat the walls. If it has been laid off and become cold, it should be worked before the trial until the walls are well heated.

VII. *The boiler and connections* should be proved to be free from leaks before beginning a test, and all water connections, including blow and extra feed pipes, should be disconnected, stopped with blank flanges, or bled through special openings beyond the valves, except the particular pipe through which water is to be fed to the boiler during the trial. During the test the blow-off and feed pipes should remain exposed to view.

If an injector is used, it should receive steam directly through a felted pipe from the boiler being tested.*

If the water is metered after it passes the injector, its temperature should be taken at the point where it leaves the injector. If the quantity is determined before it goes to the injector the temperature should be determined on the suction side of the injector, and if no change of temperature occurs other than that due to the injector, the temperature thus determined is properly that of the feed water. When the temperature changes between the injector and the boiler, as by the use of a heater or by radiation, the temperature at which the water enters and leaves the injector and that at which it enters the boiler should all be taken. In that case the weight to be used is that of the water leaving the injector, computed from the heat units if not directly measured, and the temperature that of the water entering the boiler.

Let w = weight of water entering the injector.
x = weight of steam entering the injector.
h_1 = heat units per pound of water entering injector.
h_2 = heat units per pound of steam entering injector.
h_3 = heat units per pound of water leaving injector.

* In feeding a boiler undergoing test with an injector taking steam from another boiler, or from the main steam pipe from several boilers, the evaporative results may be modified by a difference in the quality of the steam from such source compared with that supplied by the boiler being tested, and in some cases the connection to the injector may act as a drip for the main steam pipe. If it is known that the steam from the main pipe is of the same pressure and quality as that furnished by the boiler undergoing the test, the steam may be taken from such main pipe.

Then $w + x$ = weight of water leaving injector.

$$x = w \frac{h_3 - h_1}{h_2 - h_3}.$$

See that the steam main is so arranged that water of condensation cannot run back into the boiler.

VIII. *Duration of the Test.* — For tests made to ascertain either the maximum economy or the maximum capacity of a boiler, irrespective of the particular class of service for which it is regularly used, the duration should be at least 10 hours of continuous running. If the rate of combustion exceeds 25 pounds of coal per square foot of grate surface per hour, it may be stopped when a total of 250 pounds of coal has been burned per square foot of grate.

In cases where the service requires continuous running for the whole 24 hours of the day, with shifts of firemen a number of times during that period, it is well to continue the test for at least 24 hours.

When it is desired to ascertain the performance under the working conditions of practical running, whether the boiler be regularly in use 24 hours a day or only a certain number of hours out of each 24, the fires being banked the balance of the time, the duration should not be less than 24 hours.

IX. *Starting and Stopping a Test.* — The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same; the water level the same; the fire upon the grates should be the same in quantity and condition; and the walls, flues, etc., should be of the same temperature. Two methods of obtaining the desired equality of conditions of the fire may be used, viz.: those which were called in the Code of 1885 "the standard method" and "the alternate method," the latter being employed where it is inconvenient to make use of the standard method.*

X. *Standard Method of Starting and Stopping a Test.* — Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time and the

* The Committee concludes that it is best to retain the designations "standard" and "alternate," since they have become widely known and established in the minds of engineers and in the reprints of the Code of 1885. Many engineers prefer the "alternate" to the "standard" method on account of its being less liable to error due to cooling of the boiler at the beginning and end of a test.

water level* while the water is in a quiescent state, just before lighting the fire.

At the end of the test remove the whole fire, which has been burned low, clean the grates and ash pit, and note the water level when the water is in a quiescent state, and record the time of hauling the fire. The water level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and not by operating the pump after the test is completed.

XI. *Alternate Method of Starting and Stopping a Test.* — The boiler being thoroughly heated by a preliminary run, the fires are to be burned low and well cleaned. Note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the water level. Note the time, and record it as the starting time. Fresh coal which has been weighed should now be fired. The ash pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave a bed of coal on the grates of the same depth, and in the same condition, as at the start. When this stage is reached, note the time and record it as the stopping time. The water level and steam pressures should previously be brought as nearly as possible to the same point as at the start. If the water level is not the same as at the start, a correction should be made by computation, and not by operating the pump after the test is completed.

XII. *Uniformity of Conditions.* — In all trials made to ascertain maximum economy or capacity, the conditions should be maintained uniformly constant. Arrangements should be made to dispose of the steam so that the rate of evaporation may be kept the same from beginning to end. This may be accomplished in a single boiler by carrying the steam through a waste steam pipe, the discharge from which can be regulated as desired. In a battery of boilers, in which only one is tested, the draft can be regulated on the remaining boilers, leaving the test boiler to work under a constant rate of production.

Uniformity of conditions should prevail as to the pressure of steam, the height of water, the rate of evaporation, the thickness of fire, the

* The gauge glass should not be blown out within an hour before the water level is taken at the beginning and end of a test, otherwise an error in the reading of the water level may be caused by a change in the temperature and density of the water in the pipe leading from the bottom of the glass into the boiler.

times of firing and quantity of coal fired at one time, and as to the intervals between the times of cleaning the fires.

The method of firing to be carried on in such tests should be dictated by the expert or person in responsible charge of the test, and the method adopted should be adhered to by the fireman throughout the test.

XIII. *Keeping the Records.* — Take note of every event connected with the progress of the trial, however unimportant it may appear. Record the time of every occurrence and the time of taking every weight and every observation. (Appendices I, IV, V, VI, VII, VIII.)

The coal should be weighed and delivered to the fireman in equal proportions, each sufficient for not more than one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the last of each portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the test may be divided into several periods if desired, and the degree of uniformity of combustion, evaporation, and economy analyzed for each period. In addition to these records of the coal and the feed water, half-hourly observations should be made of the temperature of the feed water, of the flue gases, of the external air in the boiler room, of the temperature of the furnace when a furnace pyrometer is used, also of the pressure of steam, and of the readings of the instruments for determining the moisture in the steam. A log should be kept on properly prepared blanks containing columns for record of the various observations. (Appendix XXII.)

When the "standard method" of starting and stopping the test is used, the hourly rate of combustion and of evaporation and the horse power should be computed from the records taken during the time when the fires are in active condition. This time is somewhat less than the actual time which elapses between the beginning and end of the run. The loss of time due to kindling the fire at the beginning and burning it out at the end makes this course necessary.

XIV. *Quality of Steam.* — The percentage of moisture in the steam should be determined by the use of either a throttling or a separating steam calorimeter. The sampling nozzle should be placed in the vertical steam pipe rising from the boiler. It should be made of ½-inch

pipe, and should extend across the diameter of the steam pipe to within half an inch of the opposite side, being closed at the end and perforated with not less than twenty ⅛-inch holes equally distributed along and around its cylindrical surface, but none of these holes should be nearer than ½ inch to the inner side of the steam pipe. The calorimeter and the pipe leading to it should be well covered with felting. Whenever the indications of the throttling or separating calorimeter show that the percentage of moisture is irregular, or occasionally in excess of three per cent, the results should be checked by a steam separator placed in the steam pipe as close to the boiler as convenient, with a calorimeter in the steam pipe just beyond the outlet from the separator. The drip from the separator should be caught and weighed, and the percentage of moisture computed therefrom added to that shown by the calorimeter. (See Appendices XV to XVII.)

Superheating should be determined by means of a thermometer placed in a mercury well inserted in the steam pipe. The degree of superheating should be taken as the difference between the reading of the thermometer for superheated steam and the readings of the same thermometer for saturated steam at the same pressure as determined by a special experiment, and not by reference to steam tables.

For calculations relating to quality of steam and corrections for quality of steam, see Appendices XVIII and XIX.

XV. *Sampling the Coal and Determining its Moisture.* — As each barrow load or fresh portion of coal is taken from the coal pile, a representative shovelful is selected from it and placed in a barrel or box in a cool place and kept until the end of the trial. The samples are then mixed and broken into pieces not exceeding one inch in diameter, and reduced by the process of repeated quartering and crushing until a final sample weighing about five pounds is obtained, and the sizes of the larger pieces are such that they will pass through a sieve with ¼-inch meshes. From this sample two one-quart, air-tight glass preserving jars, or other air-tight vessels which will prevent the escape of moisture from the sample, are to be promptly filled, and these samples are to be kept for subsequent determinations of moisture and of heating value and for chemical analyses. During the process of quartering, when the sample has been reduced to about 100 pounds, a quarter to a half of it may be taken for an approximate determination of moisture. This may be made by placing it in a shallow iron pan, not over three inches deep, carefully weighing it, and setting the pan in the hottest place that can be found on the brickwork of the boiler

setting or flues, keeping it there for at least 12 hours, and then weighing it. The determination of moisture thus made is believed to be approximately accurate for anthracite and semi-bituminous coals, and also for Pittsburg or Youghiogheny coal; but it cannot be relied upon for coals mined west of Pittsburg, or for other coals containing inherent moisture. For these latter coals it is important that a more accurate method be adopted. The method recommended by the Committee for all accurate tests, whatever the character of the coal, is described as follows:

Take one of the samples contained in the glass jars and subject it to a thorough air-drying by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill adjusted so as to produce somewhat coarse grains (less than $\frac{1}{16}$ inch), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams, weigh it in a balance which will easily show a variation as small as 1 part in 1000, and dry it in an air or sand bath at a temperature between 240 and 280 degrees F. for one hour. Weigh it and record the loss, then heat and weigh it again repeatedly, at intervals of an hour or less, until the minimum weight has been reached and the weight begins to increase by oxidation of a portion of the coal. The difference between the original and the minimum weight is taken as the moisture in the air-dried coal. This moisture test should preferably be made on duplicate samples, and the results should agree within 0.3 to 0.4 of one per cent, the mean of the two determinations being taken as the correct result. The sum of the percentage of moisture thus found and the percentage of surface moisture previously determined is the total moisture. (Appendix XI.)

XVI. *Treatment of Ashes and Refuse.* — The ashes and refuse are to be weighed in a dry state. If it is found desirable to show the principal characteristics of the ash, a sample should be subjected to a proximate analysis and the actual amount of incombustible material determined. For elaborate trials a complete analysis of the ash and refuse should be made.

XVII. *Calorific Tests and Analysis of Coal.* — The quality of the fuel should be determined either by heat test or by analysis, or by both.

The rational method of determining the total heat of combustion is to burn the sample of coal in an atmosphere of oxygen gas, the coal to be sampled as directed in Article XV of this code. (See Appendices XIII and XIV.)

The chemical analysis of the coal should be made only by an expert chemist. The total heat of combustion computed from the results of the ultimate analysis may be obtained by the use of Dulong's formula (with constants modified by recent determinations), viz.: $14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right) + 4000\ S$, in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur respectively, as determined by the ultimate analysis.*

It is desirable that a proximate analysis should be made, thereby determining the relative proportions of volatile matter and fixed carbon. These proportions furnish an indication of the leading characteristics of the fuel, and serve to fix the class to which it belongs. (Appendix XII.) As an additional indication of the characteristics of the fuel, the specific gravity should be determined.

XVIII. *Analysis of Flue Gases.* — The analysis of the flue gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analyses great care should be taken to procure average samples, since the composition is apt to vary at different points of the flue. (Appendix XXXI.) The composition is also apt to vary from minute to minute, and for this reason the drawings of gas should last a considerable period of time. Where complete determinations are desired, the analyses should be intrusted to an expert chemist. For approximate determinations the Orsat† or the Hempel ‡ apparatus may be used by the engineer. (See Appendix XXXIII.)

For the continuous indication of the amount of carbonic acid present in the flue gases, an instrument may be employed which shows the weight of the sample of gas passing through it. (Appendix XXXIX.)

XIX. *Smoke Observations.* — It is desirable to have a uniform system of determining and recording the quantity of smoke produced where bituminous coal is used. The system commonly employed is to express the degree of smokiness by means of percentages dependent upon the judgment of the observer. The Committee does not place much value

* Favre and Silberman give 14,544 B.T.U. per pound carbon; Berthelot 14,647 B.T.U. Favre and Silberman give 62,032 B.T.U. per pound hydrogen; Thomsen 61,816 B.T.U.

† See R. S. Hale's paper on "Flue Gas Analysis," *Transactions*, Vol. XVIII, p. 901.

‡ See Hempel on "Gas Analysis."

upon a percentage method, because it depends so largely upon the personal element, but if this method is used, it is desirable that, so far as possible, a definition be given in explicit terms as to the basis and method employed in arriving at the percentage. The actual measurement of a sample of soot and smoke by some form of meter is to be preferred. (See Appendices XXXIV and XXXV.)

XX. *Miscellaneous.* — In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are in general unnecessary for ordinary tests. These are the measurement of the air supply, the determination of its contained moisture, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

As these determinations are rarely undertaken, it is not deemed advisable to give directions for making them.

XXI. *Calculations of Efficiency.* — Two methods of defining and calculating the efficiency of a boiler are recommended. They are:

$$1.\quad \text{Efficiency of the boiler} = \frac{\text{Heat absorbed per lb. combustible}}{\text{Calorific value of 1 lb. combustible}}.$$

$$2.\quad \text{Efficiency of the boiler and grate} = \frac{\text{Heat absorbed per lb. coal}}{\text{Calorific value of 1 lb. coal}}.$$

The first of these is sometimes called the efficiency based on combustible, and the second the efficiency based on coal. The first is recommended as a standard of comparison for all tests, and this is the one which is understood to be referred to when the word "efficiency" alone is used without qualification. The second, however, should be included in a report of a test, together with the first, whenever the object of the test is to determine the efficiency of the boiler and furnace together with the grate (or mechanical stoker), or to compare different furnaces, grates, fuels, or methods of firing.

The heat absorbed per pound of combustible (or per pound coal) is to be calculated by multiplying the equivalent evaporation from and at 212 degrees per pound combustible (or coal) by 965.7 (Appendix XX.)

XXII. *The Heat Balance.* — An approximate "heat balance," or statement of the distribution of the heating value of the coal among the several items of heat utilized and heat lost, may be included in the

report of a test when analyses of the fuel and of the chimney gases have been made. It should be reported in the following form:

HEAT BALANCE, OR DISTRIBUTION OF THE HEATING VALUE OF THE COMBUSTIBLE.

Total Heat Value of 1 pound of combustible..........B.T.U.

	B.T.U.	Per Cent.
1. Heat absorbed by the boiler = evaporation from and at 212 degrees per pound of combustible $\times$ 965.7.		
2. Loss due to moisture in coal = per cent of moisture referred to combustible $\div$ 100 $\times$ $[(212 - t) + 966 + 0.48\,(T - 212)]$ (t = temperature of air in the boiler room, T = that of the flue gases).		
3. Loss due to moisture formed by the burning of hydrogen = per cent of hydrogen to combustible $\div$ 100 $\times$ 9 $\times$ $[(212 - t) + 966 + 0.48\,(T - 212)]$.		
4. * Loss due to heat carried away in the dry chimney gases = weight of gas per pound of combustible $\times$ 0.24 $\times$ $(T - t)$.		
5. † Loss due to incomplete combustion of carbon = $\dfrac{CO}{CO_2 + CO} \times \dfrac{\text{per cent } C \text{ in combustible}}{100} \times 10{,}150$.		
6. Loss due to unconsumed hydrogen and hydrocarbons, to heating the moisture in the air, to radiation, and unaccounted for. (Some of these losses may be separately itemised if data are obtained from which they may be calculated.)		
Totals....................................		100.00

* The weight of gas per pound of carbon burned may be calculated from the gas analyses as follows:

Dry gas per pound carbon $= \dfrac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)}$, in which CO_2, CO, O, and N are the percentages by volume of the several gases. As the sampling and analyses of the gases in the present state of the art are liable to considerable errors, the result of this calculation is usually only an approximate one. The heat balance itself is also only approximate for this reason, as well as for the fact that it is not possible to determine accurately the percentage of unburned hydrogen or hydrocarbons in the flue gases. (See Appendix XXXII.)

The weight of dry gas per pound of combustible is found by multiplying the dry gas per pound of carbon by the percentage of carbon in the combustible and dividing by 100.

† CO_2 and CO are respectively the percentage by volume of carbonic acid and carbonic oxide in the flue gases. The quantity 10,150 = No. heat units generated by burning to carbonic acid one pound of carbon contained in carbonic oxide.

XXIII. *Report of the Trial.* — The data and results should be reported in the manner given in either one of the two following tables, omitting lines where the tests have not been made as elaborately as provided for in such tables. Additional lines may be added for data relating to the specific object of the test. The extra lines should be

classified under the headings provided in the tables, and numbered as per preceding line, with sub letters *a*, *b*, etc. The Short Form of Report, Table No. 2, is recommended for commercial tests and as a convenient form of abridging the longer form for publication when saving of space is desirable. For elaborate trials, it is recommended that the full log of the trial be shown graphically, by means of a chart. (Appendix XXXVIII.)

TABLE NO. 1.

DATA AND RESULTS OF EVAPORATIVE TEST,

Arranged in accordance with the Complete Form advised by the Boiler Test Committee of the American Society of Mechanical Engineers. Code of 1899.

Made by .... of .... boiler at .... to determine ....

....

Principal conditions governing the trial ....

....

....

Kind of fuel * ....

Kind of furnace ....

State of the weather ....

Method of starting and stopping the test ("standard" or "alternate," Art. X and XI, Code) ....

1. *Date of trial* ....
2. *Duration of trial* .... hours.

Dimensions and Proportions.

A complete description of the boiler, and drawings of the same if of unusual type, should be given on an annexed sheet. (See Appendix X.)

3. *Grate surface* .... *width* .... *length* .... *area* .... square feet
4. Height of furnace .... inches.
5. Approximate width of air spaces in grate .... inch.
6. Proportion of air space to whole grate surface .... per cent.
7. *Water-heating surface* .... square feet.
8. *Superheating surface* .... "
9. Ratio of water-heating surface to grate surface .... — to 1.
10. Ratio of minimum draft area to grate surface .... 1 to —.

Average Pressures.

11. *Steam pressure by gauge* .... lb. per sq. in.
12. *Force of draft between damper and boiler* .... in. of water.
13. Force of draft in furnace .... " "
14. Force of draft or blast in ash pit .... " "

* The items printed in italics correspond to the items in the "Short Form of Code."

Average Temperatures.

15. Of external air..	degrees.
16. Of fire room..	"
17. Of steam..	"
18. Of feed water entering heater..................................	"
19. Of feed water entering economizer..............................	"
20. *Of feed water entering boiler*................................	"
21. *Of escaping gases from boiler*................................	"
22. Of escaping gases from economizer..............................	"

Fuel.

23. Size and condition..	
24. Weight of wood used in lighting fire............................	pounds.
25. *Weight of coal as fired* *.....................................	"
26. *Percentage of moisture in coal* †..............................	per cent.
27. *Total weight of dry coal consumed*.............................	pounds.
28. *Total ash and refuse*..	"
29. Quality of ash and refuse.......................................	
30. Total combustible consumed......................................	pounds.
31. *Percentage of ash and refuse in dry coal*......................	per cent.

Proximate Analysis of Coal.

(App. XII.)

	Of Coal.	Of Combustible.
32. Fixed carbon..............................	per cent.	per cent.
33 Volatile matter............................	"	"
34. Moisture..................................	"	——
35. Ash.......................................	"	——
	100 per cent.	100 per cent.
36. Sulphur, separately determined............	"	"

Ultimate Analysis of Dry Coal.

(Art. XVII, Code.)

	Of Coal.	Of Combustible.
37. Carbon (C)................................	per cent.	per cent.
38. Hydrogen (H)..............................	"	"
39. Oxygen (O)................................	"	"
40. Nitrogen (N)..............................	"	"
41. Sulphur (S)...............................	"	"
42. Ash.......................................	"	——
	100 per cent.	100 per cent.
43. Moisture in sample of coal as received....	"	"

* Including equivalent of wood used in lighting the fire, not including unburned coal withdrawn from furnace at times of cleaning and at end of test. One pound of wood is taken to be equal to 0.4 pound of coal, or, in case greater accuracy is desired, as having a heat value equivalent to the evaporation of 6 pounds of water from and at 212 degrees per pound. ($6 \times 965.7 = 5794$ B.T.U.)

† This is the total moisture in the coal as found by drying it artificially, as described in Art. XV of Code.

Analysis of Ash and Refuse.

44. Carbon..	per cent.
45. Earthy matter..	"

Fuel per Hour.

46. *Dry coal consumed per hour*..................................	pounds.
47. Combustible consumed per hour..............................	"
48. *Dry coal per square foot of grate surface per hour*.................	"
49. Combustible per square foot of water-heating surface per hour...	"

Calorific Value of Fuel.

(Art. XVII, Code.)

50. *Calorific value by oxygen calorimeter, per pound of dry coal*........	B.T.U.
51. *Calorific value by oxygen calorimeter, per pound of combustible*.....	"
52. Calorific value by analysis, per pound of dry coal *.............	"
53. Calorific value by analysis, per pound of combustible............	"

Quality of Steam.

(App. XV to XIX.)

54. *Percentage of moisture in steam*..................................	per cent.
55. *Number of degrees of superheating*.............................	degrees.
56. Quality of steam (dry steam = unity). (For exact determination of the factor of correction for quality of steam see Appendix XVIII)..	

Water.

(App. I, IV, VII, VIII.)

57. *Total weight of water fed to boiler* †............................	pounds.
58. Equivalent water fed to boiler from and at 212 degrees..........	"
59. *Water actually evaporated, corrected for quality of steam*...........	"
60. Factor of evaporation ‡....................................	pounds.
61. Equivalent water evaporated into dry steam from and at 212 degrees. (Item 59 × Item 60.)............................	"

Water per Hour.

62. *Water evaporated per hour, corrected for quality of steam*..........	"
63. *Equivalent evaporation per hour from and at 212 degrees*..........	"
64. *Equivalent evaporation per hour from and at 212 degrees per square foot of water-heating surface*............................	"

* See formula for calorific value under Article XVII of Code.

† Corrected for inequality of water level and of steam pressure at beginning and end of test.

‡ Factor of evaporation $= \frac{H-h}{965.7}$ in which H and h are respectively the total heat in steam of the average observed pressure, and in water of the average observed temperature of the feed.

Horse Power.

65. *Horse power developed. (34½ pounds of water evaporated per hour into dry steam from and at 212 degrees equals one horse power.)** ... horse power.
66. *Builders' rated horse power* .... "
67. *Percentage of builders' rated horse power developed* .... per cent.

Economic Results.

68. *Water apparently evaporated per pound of coal under actual conditions. (Item 58 ÷ Item 25.)* .... pounds.
69. *Equivalent evaporation from and at 212 degrees per pound of wet coal. (Item 61 ÷ Item 25.)* .... "
70. *Equivalent evaporation from and at 212 degrees per pound of dry coal. (Item 61 ÷ Item 27.)* .... "
71. *Equivalent evaporation from and at 212 degrees per pound of combustible. (Item 61 ÷ Item 30.)* .... "

(If the equivalent evaporation, Items 69, 70, and 71, is not corrected for the quality of steam, the fact should be stated.)

Efficiency.

(See Art. XXI, Code.)

72. *Efficiency of the boiler; heat absorbed by the boiler per pound of combustible divided by the heat value of one pound of combustible* † ... per cent.
73. *Efficiency of boiler, including the grate; heat absorbed by the boiler, per pound of dry coal fired, divided by the heat value of one pound of dry coal* ‡ .... "

Cost of Evaporation.

74. *Cost of coal per ton of —— pounds delivered in boiler room* .... $
75. Cost of fuel for evaporating 1000 pounds of water under observed conditions .... $
76. *Cost of fuel used for evaporating* 1000 *pounds of water from and at* 212 *degrees* .... $

Smoke Observations.

(App. XXXIV and XXXV.)

77. Percentage of smoke as observed .... per cent.
78. Weight of soot per hour obtained from smoke meter .... ounces.
79. Volume of soot per hour obtained from smoke meter .... cubic inches.

* Held to be the equivalent of 30 pounds of water per hour evaporated from 100 degrees F. into dry steam at 70 pounds gauge pressure. (See Introduction to Code.)

† In all cases where the word combustible is used, it means the coal without moisture and ash, but including all other constituents. It is the same as what is called in Europe "coal dry and free from ash."

‡ The heat value of the coal is to be determined either by an oxygen calorimeter or by calculation from ultimate analysis.

Methods of Firing.

80. Kind of firing (spreading, alternate, or coking)................
81. Average thickness of fire.................................
82. Average intervals between firings for each furnace during time when fires are in normal condition........................
83. Average interval between times of levelling or breaking up.......

Analyses of the Dry Gases.

84. Carbon dioxide (CO_2)................................	per cent
85. Oxygen (O)...	"
86. Carbon monoxide (CO)...................................	"
87. Hydrogen and hydrocarbons..............................	"
88. Nitrogen (by difference) (N)...........................	"
	100 per cent.

TABLE NO. 2.

DATA AND RESULTS OF EVAPORATIVE TEST,

Arranged in accordance with the Short Form advised by the Boiler Test Committee of the American Society of Mechanical Engineers. Code of 1899.

Made by..................on...............boiler, at..................to determine..
Kind of fuel...
Kind of furnace..
Method of starting and stopping the test ("standard" or "alternate," Art. X and XI, Code) ....

Grate surface..	square feet.
Water-heating surface..	"
Superheating surface...	"

Total Quantities.

1. Date of trial...	
2. Duration of trial...	hours.
3. Weight of coal as fired *.....................................	pounds.
4. Percentage of moisture in coal *..............................	per cent.
5. Total weight of dry coal consumed.............................	pounds.
6. Total ash and refuse..	"
7. Percentage of ash and refuse in dry coal......................	per cent.
8. Total weight of water fed to the boiler.......................	pounds.
9. Water actually evaporated, corrected for moisture or superheat in steam..	"
10. Equivalent water evaporated into dry steam from and at 212 degrees..	"

* See foot-notes of Complete Form.

Hourly Quantities.

11. Dry coal consumed per hour	pounds.
12. Dry coal per hour per square foot of grate surface	"
13. Water fed per hour	"
14. Equivalent water evaporated per hour from and at 212 degrees corrected for quality of steam	"
15. Equivalent water evaporated per hour per square foot of water-heating surface	"

Average Pressures, Temperatures, etc.

16. Average boiler pressure	lb. per sq. in.
17. Average temperature of feed water	degrees.
18. Average temperature of escaping gases	"
19. Average force of draft between damper and boiler	in. of water.
20. Percentage of moisture in steam, or number of degrees of superheating	

Horse Power.

21. Horse power developed (Item 14 ÷ 34½)	horse power.
22. Builders' rated horse power	"
23. Percentage of builders' rated horse power developed	per cent.

Economic Results.

24. Water apparently evaporated per pound of coal under actual conditions. (Item 8 ÷ Item 3.)	pounds.
25. Equivalent water actually evaporated from and at 212 degrees per pound of wet coal. (Item 9 ÷ Item 3.)	"
26. Equivalent evaporation from and at 212 degrees per pound of dry coal. (Item 9 ÷ Item 5.)	"
27. Equivalent evaporation from and at 212 degrees per pound of combustible. [Item 9 ÷ (Item 5 − Item 6).]	"

(If Items 25, 26, and 27 are not corrected for quality of steam, the fact should be stated.)

Efficiency.

28. Heating value of the coal per pound	B.T.U.
29. Heating value of the combustible per pound	"
30. Efficiency of boiler (based on combustible)	per cent.
31. Efficiency of boiler, including grate (based on coal)	"

Cost of Evaporation.

32. Cost of coal per ton delivered in boiler room	$
33. Cost of coal required for evaporation of 1000 pounds of water from and at 212 degrees	$

LIST OF APPENDICES TO CODE.*

No. of Appendix.

* Only a few of the appendices are reprinted.

† Contributed by members of the Society and accepted by the Committee for publication in the Appendix.

APPENDIX XX.

Efficiency of the Boiler.

The efficiency of the boiler, including the grate, or the efficiency based on coal, is the quotient arising from dividing the heat absorbed by the boiler by the heating value of the total amount of coal supplied to the boiler, including the coal which falls through the grate. It may be conveniently calculated by multiplying the number of pounds of water evaporated from and at 212 degrees F. into dry steam per pound of dry coal by 965.7 and dividing the product by the heating value in British thermal units of one pound of dry coal.

The efficiency of the boiler, not including the grate, or the efficiency based on combustible, is the quotient arising from dividing the heat absorbed by the boiler by the heating value of the combustible burned. It may be calculated by multiplying the number of pounds of water evaporated from and at 212 degrees F. into dry steam per pound of combustible by 965.7 and dividing the product by the heating value in British thermal units of one pound of combustible; the term "combustible" being defined as coal dry and free from ash, or the coal supplied to the boiler less its moisture and the ash and unburned coal which falls through the grate or is otherwise withdrawn from the furnace.

The efficiency of the boiler, not including the grate (or the efficiency based upon combustible), is a more accurate measure of comparison of different boilers than the efficiency including the grate (or the efficiency based upon coal), for the latter is subject to a number of variable conditions, such as size and character of the coal, air spaces between the grate bars, skill of the fireman in saving coal from falling through the grate, etc. It is, moreover, subject to errors of sampling the coal for drying and for analysis, which affect the result to a greater degree than they do the efficiency based upon combustible, for the reason that the heating value of one pound of combustible of any sample selected from a given lot, such as a car load, of coal is practically a constant quantity and is independent of the percentage of moisture and ash in the sample; while the sample itself, upon the heating value of which the efficiency based on coal is calculated, may differ in its percentage of moisture and ash from the average coal used in the boiler test.

When the object of a boiler test is to determine its efficiency as an absorber of heat, or to compare it with other boilers, the efficiency based on combustible is the one which should be used; but when the object of the test is to determine the efficiency of the combination of the boiler, the furnace, and the grate, the efficiency based on coal must necessarily be used.

It has been proposed that in reporting the efficiency of a boiler when the fuel used contains hydrogen, the efficiency should be considered to be the sum of the percentage of the heating value of the fuel which is utilized by the boiler in making steam and of the percentage of that heating value which is lost in the shape of latent heat in the moisture in the chimney gases, which moisture is formed by the burning of the hydrogen. This latent heat may amount to over three per cent of the total heating value of the fuel. The reason assigned for this proposal is that, since it is impossible for this heat to be utilized by the boiler because the gases are discharged at a temperature above 212 degrees F. it should not be charged against the boiler. The writer does not consider it advisable that this method of reporting the efficiency should be adopted (1) because it is opposed to the generally accepted definition of efficiency, which is the useful work received from an apparatus divided by the work (or heat value of the fuel) put into it; (2) because in order to calculate it it is necessary to know both the percentage of hydrogen in the coal and whether or not all of this hydrogen has been burned to H_2O, the first requiring an analysis of the coal, which is not always obtainable, and the second an analysis of the gases for hydrogen, which cannot be obtained with any approximation to accuracy with our present methods of sampling and analyzing gases; and (3) because it is opposed to the almost universal custom in reporting boiler tests. It is true that the latent heat of the H_2O in the chimney gases cannot be utilized (unless an economizer which discharges its gases below 212 degrees is used), and it is not the fault of the boiler that it cannot be utilized. It may be considered the misfortune of the boiler, when tested with hydrogenous coal, similar to the misfortune under which an engine labors when it is tested while supplied with a condenser which gives a vacuum of less than 30 inches of mercury. The engine might give a higher efficiency with a vacuum of 30 inches than it would with one of 27 or 28 inches; but it is not customary to credit the engine with the efficiency which it loses on account of the imperfect vacuum.

Since it is well understood that a boiler cannot show quite as high an efficiency (as commonly defined) when using bituminous coal high in hydrogen as when using anthracite nearly free from hydrogen, no harm is done, and much confusion is avoided, by reporting the efficiency as the percentage of the heating value of the coal which is actually utilized in making steam. The fact that bituminous coal is used is always stated in the report of a test made with that coal. If desired a statement may also be made in the "heat balance" of the approximate or estimated percentage of heat which is lost in the latent heat of the moisture in the chimney gases, together with the loss due to moisture in the coal.

APPENDIX XXV.

Discrepancy between Commercial and Experimental Results.

The final result sought by manufacturers, in initiating tests of steam or other machinery in actual use, is the value of the work done measured in dollars and cents. In some cases the broad question is raised as to the saving that may be accomplished by installing improved boilers, engines, or other machinery; but more generally it is desired to ascertain what can be done to produce saving with the apparatus already in place under the actual conditions that prevail at the particular location. In both these cases it is necessary to ascertain the average cost of the work done commercially previous to the test. Frequently, in fact generally, this important fact will not be ascertained by an elaborate trial, for the reason that everything will be put in order for the test, and all details of the trial be conducted so carefully that the losses due to average carelessness or want of skill in the past will be eliminated, the engineer making the test will not receive proper credit, and the owners on seeing the report may conclude that they are already doing very well, and perhaps continue old methods with fancied security. If the cost of the output of the factory for a given time were ascertained in terms of the coal burned during the same time, and compared with the corresponding cost for the time of the trial, the latter would frequently be found to be one-eighth to one-third less than the former, and it might not be possible to tell what had caused the difference; for instance, whether it was due to putting in order the machinery prior to the tests, to greater care exercised by the fireman under the spur of careful watching, or whether, as is usually claimed, the coal was different, etc., etc. The losses are generally due in the main to the carelessness of the firemen. It follows, therefore, that the cost of the power under average conditions must be obtained in some quiet way preliminarily. Frequently the comparison of the output of the factory with the coal burned will not be sufficiently accurate, and it will be necessary to devise some corresponding check which will not interfere with the regular routine of the establishment. The work of the boilers may be checked by arranging a meter so as to continuously measure the feed water; and its record, compared with the total weight of coal *purchased*, will frequently give the check desired. Such a check becomes more difficult when it is desirable to ascertain the performances of particular boilers, and the coal supply is common to all boilers; but by assigning particular weighed car loads of coal to the particular boilers, without any intimation to the firemen that they are being watched, it may be

possible to ascertain the average performance of the boilers used for the particular purpose. Preliminary experiments of this kind conducted without notice to employees, and continued through a long period, will furnish a basis for comparison with elaborate tests, and it will then be possible to point out clearly where the several losses have taken place, and the testing engineer will get the credit for the saving shown.

C. E. E.

APPENDIX XXXV.

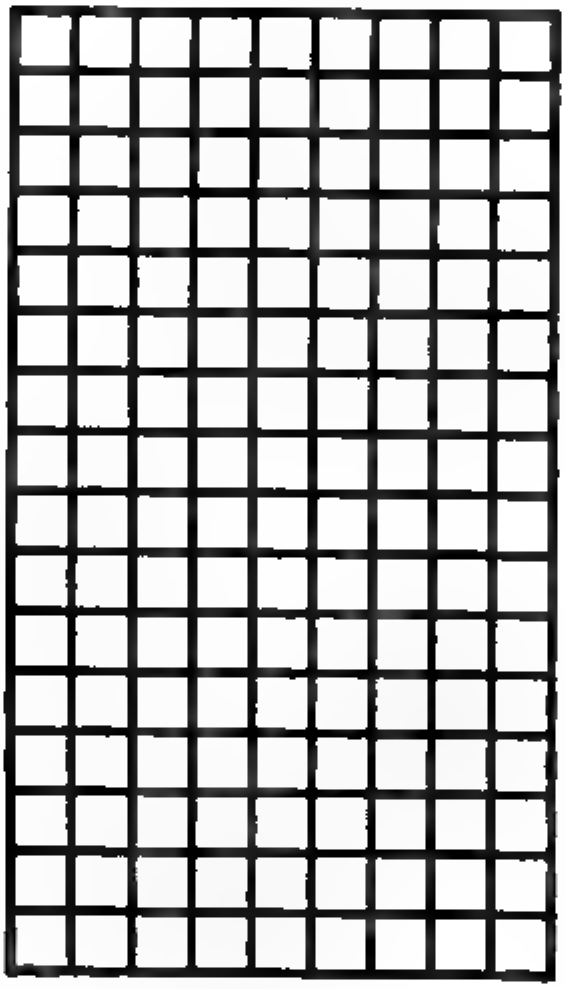
The Ringelmann Smoke Chart.

Professor Ringelmann, of Paris, has invented a system of determining the relative density or blackness of smoke, which has been communicated to the writer by Mr. Bryan Donkin, of London, and published in *Engineering News* of November 11, 1897. In making observations of the smoke proceeding from a chimney four cards ruled like those in the cut, together with a card printed in solid black and another left entirely white, are placed in a horizontal row and hung at a point about 50 feet from the observer and as nearly as convenient in line with the chimney. At this distance the lines become invisible, and the cards appear to be of different shades of gray, ranging from very light gray to almost black. The observer glances from the smoke coming from the chimney to the cards, which are numbered from 0 to 5, determines which card most nearly corresponds with the color of the smoke and makes a record accordingly, noting the time. Observations should be made continuously during say one minute, and the estimated average density during that minute recorded, and so on, records being made once every minute. The average of all the records made during a boiler test is taken as the average figure for the smoke density during the test, and the whole of the record is plotted on cross-section paper in order to show how the smoke varied in density from time to time. A rule by which the cards may be reproduced is given by Professor Ringelmann as follows:

Card 0 — All white.
Card 1 — Black lines 1 mm. thick, 10 mm. apart, leaving spaces 9 mm. square.
Card 2 — Lines 2.3 mm. thick, spaces 7.7 mm. square.
Card 3 — Lines 3.7 mm. thick, spaces 6.3 mm. square.
Card 4 — Lines 5.5 mm. thick, spaces 4.5 mm. square.
Card 5 — All black.

The cards as printed in Fig. 452 are much smaller than those used by Professor Ringelmann, but the thickness and the spacing of the lines are the same.

W. K.



No.1

No.2

No.3

No.4

FIG. 462. Ringelman Smoke Chart.

APPENDIX XXXVIII.

Chart Showing Graphically the Log of a Trial.

The well-known method of plotting observations and data on cross-section paper and making a chart applying to the test is a useful means of representing the exact uniformity of conditions existing

FIG. 453.

during a trial. Such a chart is illustrated in Fig. 453, in which the abscissæ represent times and the ordinates, on appropriate scales, the various observations and data. G. H. B.

APPENDIX C.

RULES FOR CONDUCTING STEAM-ENGINE TESTS.*

Code of 1902.

I. *Object of Test.* — Ascertain at the outset the specific object of the test, whether it be to determine the fulfillment of a contract guarantee, to ascertain the highest economy obtainable, to find the working economy and defects under conditions as they exist, to ascertain the performance under special conditions, to determine the effect of changes in the conditions, or to find the performance of the entire boiler and engine plant, and prepare for the test accordingly.

II. *General Condition of the Plant.* — Examine the engine and the entire plant concerned in the test; note its general condition and any points of design, construction, or operation which bear on the objects in view. Make a special examination of the valves and pistons for leakage by applying the working pressures with the engine at rest, and observe the quantity of steam, if any, blowing through per hour.

If the trial has for an object the determination of the highest efficiency obtainable, the valves and pistons must first be made tight, and all parts of the engine and its auxiliaries, and all other parts of the plant concerned, should be put in the best possible working condition.

III. *Dimensions, etc.* — Measure or check the dimensions of the cylinders in any case, this being done when they are hot. If they are much worn, the average diameter should be determined. Measure also the clearance, which should be done, if possible, by filling the spaces with water previously measured, the piston being placed at the end of the stroke. If the clearance cannot be measured directly, it can be determined approximately from the working drawings of the cylinder.

Measure also the dimensions of auxiliaries and accessories, also those of the boilers so far as concerned in attaining the objects. It is well to supplement these determinations with a sketch or sketches showing the general features and arrangement of the different parts of the plant.

* From the final report of the committee appointed to standardize a system of testing steam engines. Trans. A.S.M.E., Vol. XXIV. Greatly abridged.

IV. *Coal.* — When the trial involves the complete plant, embracing boilers as well as engine, determine the character of coal to be used. The class, name of the mine, size, moisture, and quality of the coal should be stated in the report. It is desirable, for purposes of comparison, that the coal should be of some recognized standard quality for the locality where the plant is situated.

V. *Calibration of Instruments.* — All instruments and apparatus should be calibrated and their reliability and accuracy verified by comparison with recognized standards. Such apparatus as is liable to change or become broken during a test, as gauges, indicator springs, and thermometers, should be calibrated before and after the test. The accuracy of scales should be verified by standard weights. When a water meter is used, special attention should be given to its calibration, verifying it both before and after the trial, and, if possible, during its progress, the conditions in regard to water pressure and rate of flow being made the same in the calibrations as exist throughout the trial.

VI. *Leakages of Steam, Water, etc.* — In all tests except those of a complete plant made under conditions as they exist, the boiler and its connections, both steam and feed, as also the steam piping leading to the engine and its connections, should, so far as possible, be made tight. If absolute tightness cannot be obtained (in point of fact it rarely can be), proper allowance should be made for such leakage in determining the steam actually consumed by the engine. This, however, is not required where a surface condenser is used and the water consumption is determined by measuring the discharge of the air pump. In such cases it is necessary to make sure that the condenser is tight, both before and after the test, against the entrance of circulating water, or if such occurs to make proper correction for it, determining it under the working difference of pressure. Should there be excessive leakage of the condenser it should be remedied before the test is made. When the steam consumption is determined by measuring the discharge of the air pump, any leakage about the valve or piston rods of the engine should be carefully guarded against.

Make sure that there is no leakage at any of the connections with the apparatus provided for measuring and supplying the feed water which could affect the results. All connections should, so far as possible, be visible and be blanked off, and where this cannot be done, satisfactory assurance should be obtained that there is no leakage either in or out.

VII. *Duration of Test.* — The duration of a test should depend largely upon its character and the objects in view. The standard heat test of an engine, and, likewise, a test for the simple determination of the feed-water consumption, should be continued for at least five hours, unless the class of service precludes a continuous run of so long duration. It is desirable to prolong the test the number of hours stated to obtain a number of consecutive hourly records as a guide in analyzing the reliability of the whole.

Where the water discharged from the surface condenser is measured for successive short intervals of time, and the rate is found to be uniform, the test may be of a much shorter duration than where the feed water is measured to the boiler. The longer the test with a given set of conditions the more accurate the work, and no test should be so short that it cannot be divided into several intervals which will give results agreeing substantially with each other.

The commercial test of a complete plant, embracing boilers as well as engine, should continue at least one full day of twenty-four hours, whether the engine is in motion during the entire time or not. A continuous coal test of a boiler and engine should be of at least ten hours' duration, or the nearest multiple of the interval between times of cleaning fires.

VIII. *Starting and Stopping a Test.* — (*a*) *Standard Heat Test and Feed-Water Test of Engine:* The engine having been brought to the normal condition of running, and operated a sufficient length of time to be thoroughly heated in all its parts, and the measuring apparatus having been adjusted and set to work, the height of water in the gauge glasses of the boilers is observed, the depth of water in the reservoir from which the feed water is supplied is noted, the exact time of day is observed, and the test held to commence. Thereafter the measurements determined upon for the test are begun and carried forward until its close. If practicable, the test may be commenced at some even hour or minute, but it is of the first importance to begin at such time as reliable observations of the water heights are obtained, whatever the exact time happens to be when these are satisfactorily determined. When the time for the close of the test arrives, the water should, if possible, be brought to the same height in the glasses and to the same depth in the feed-water reservoir as at the beginning, delaying the conclusion of the test if necessary to bring about this similarity of conditions. If differences occur, the proper corrections must be made.

(*b*) *Complete Engine and Boiler Test:* For a continuous running test of combined engine or engines, and boiler or boilers, the same

directions apply for beginning and ending the feed-water measurements as those just referred to under Section (*a*). The time of beginning and ending such a test should be the regular time of cleaning the fires, and the exact time of beginning and ending should be the time when the fires are fully cleaned, just preparatory to putting on fresh coal. In cases where there are a number of boilers, and it is inconvenient or undesirable to clean all fires at once, the time of beginning the test should be deferred until they are all cleaned and in a satisfactory state, all the fires being then burned down to a uniformly thin condition, the thickness and condition being estimated and the test begun just before firing the new coal previously weighed. The ending of the test is likewise deferred until the fires are all satisfactorily cleaned, being again burned down to the same uniformly thin condition as before, and the time of closing being taken just before replenishing the fires with new coal.

For a commercial test of a combined engine and boiler, whether the engine runs continuously for the full twenty-four hours of the day or only a portion of the time, the fires in the boilers being banked during the time when the engine is not in motion, the beginning and ending of the test should occur at the regular time of cleaning the fires, the method followed being that already given. In cases where the engine is not in continuous motion, as, for example, in textile mills, where the working time is ten or eleven hours out of the twenty-four, and the fires are cleaned and banked at the close of the day's work, the best time for starting and stopping a test is the time just before banking, when the fires are well burned down and the thickness and condition can be most satisfactorily judged. In these, as in all other cases noted, the test should be begun by observing the exact time, the thickness and condition of the fires on the grates, the height of water in the gauge glasses of the boilers, the depth of the water in the reservoir from which the feed water is supplied, and other conditions relating to the trial, the same observations being again taken at the end of the test, and the conditions in all respects being made as nearly as possible the same as at the beginning.

IX. *Measurement of Heat Units consumed by the Engine.* — The measurement of the heat consumption requires the measurement of each supply of feed water to the boiler — that is, the water supplied by the main feed pump, that supplied by auxiliary pumps, such as jacket water, water from separators, drips, etc., and water supplied by gravity or other means; also the determination of the temperature of the water supplied from each source, together with the pressure and quality of the steam.

The temperatures at the various points should be those applying to

the working conditions. The temperature of the feed water should be taken near the boiler. This causes the engine to suffer a disadvantage from the heat lost by radiation from the pipes which carry the water to the boiler, but it is, nevertheless, advisable on the score of simplicity. Such pipes would, therefore, be considered a portion of the engine plant. This conforms with the rule already recommended for the tests of pumping engines where the duty per million heat units is computed from the temperature of the feed water taken near the boiler. It frequently happens that the measurement of the water requires a change in the usual temperature of supply. For example, where the main supply is ordinarily drawn from a hot well in which the temperature is say 100 degrees F., it may be necessary, owing to the low level of the well, to take the supply from some source under a pressure or head sufficient to fill the weighing tanks used, and this supply may have a temperature much below that of the hot well; possibly as low as 40 degrees F. The temperature to be used is not the temperature of the water as weighed in this case, but that of the working temperature of the hot well. The working temperature in cases like this must be determined by a special test, and included in the log sheets.

The heat to be determined is that used by the entire engine equipment, embracing the main cylinders and all auxiliary cylinders and mechanism concerned in the operation of the engine, including the air pump, circulating pump, and feed pumps, also the jacket and reheater when these are used. No deduction is to be made for steam used by auxiliaries unless these are shown by test to be unduly wasteful. In this matter an exception should be made in cases of guarantee tests where the engine contractor furnishes all the auxiliaries referred to. He should, in that case, be responsible for the whole, and no allowance should be made for inferior economy, if such exists. Should a deduction be made on account of the auxiliaries being unduly wasteful, the method of waste and its extent, as compared with the wastes of the main engine or other standard of known value, shall be reported definitely.

The steam pressure and the quality of the steam are to be taken at some point conveniently near the throttle valve. The quantity of steam used by the calorimeter must be determined and properly allowed for. (See Article XVI, on " Quality of Steam.")

X. *Measurement of Feed Water or Steam Consumption of Engine, etc.* — The method of determining the steam consumption applicable to all plants is to measure all the feed water supplied to the boilers, and deduct therefrom the water discharged by separators and drips, as also the water and steam which escape on account of leakage of the boiler and its pipe connections and leakage of the steam main and branches

connecting the boiler and the engine. In plants where the engine exhausts into a surface condenser the steam consumption can be measured by determining the quantity of water discharged by the air pump, corrected for any leakage of the condenser, and adding thereto the steam used by jackets, reheaters, and auxiliaries as determined independently. If the leakage of the condenser is too large to satisfactorily allow for it, the condenser should, of course, be repaired and the leakage again determined before making the test.

In measuring the water it is best to carry it through a tank or tanks resting on platform weighing scales suitably arranged for the purpose, the water being afterwards emptied into a reservoir beneath, from which the pump is supplied.

Where extremely large quantities of water must be measured, or in some places relatively small quantities, the orifice method of measuring is one that can be applied with satisfactory results. In this case the average head of water on the orifice must be determined, and, furthermore, it is important that means should be at hand for calibrating the discharge of the orifice under the conditions of use.

The corrections or deductions to be made for leakage above referred to should be applied only to the standard heat-unit test and tests for determining simply the steam or feed-water consumption, and not to coal tests of combined engine and boiler equipment. In the latter, no correction should be made except for leakage of valves connecting to other engines and boilers, or for steam used for purposes other than the operation of the plant under test. Losses of heat due to imperfections of the plant should be charged to the plant, and only such losses as are concerned in the working of the engine alone should be charged to the engine.

In measuring jacket water or any supply under pressure which has a temperature exceeding 212 degrees F., the water should first be cooled, as may be done by discharging it into a tank of cold water previously weighed, or by passing it through a coil of pipe submerged in running and colder water, preventing thereby the loss of evaporation which occurs when such hot water is discharged into the open air.

XI. *Measurement of Steam used by Auxiliaries.* — Although the steam used by the auxiliaries — embracing the air pump, circulating pump, feed pump, and any other apparatus of this nature, supposing them to be steam-driven, also the steam jackets, reheaters, etc., which consume steam required for the operation of the engine — is all included in the measurement of the steam consumption, as pointed out in Article X, yet it is highly desirable that the quantity of steam

used by the auxiliaries, and in many cases that used by each auxiliary, should be determined exactly, so that the net consumption of the main engine cylinders may be ascertained and a complete analysis made of the entire work of the engine plant. Where the auxiliary cylinders are non-condensing, the steam consumption can often be measured by carrying the exhaust for the purpose into a tank of cold water resting on scales or through a coil of pipe surrounded by cold running water. Another method is to run the auxiliaries as a whole, or one by one, from a spare boiler (preferably a small vertical one), and measure the feed water supplied to this boiler. The steam used by the air and circulating pumps may be measured by running them under, as near as possible, the working conditions and speed, the main engine and other auxiliaries being stopped, and testing the consumption by the measuring apparatus used on the main trial. For a short trial, to obtain approximate results, measurement can be made by the water-gauge glass method, the feed supply being shut off. When the engine has a surface condenser, the quantity of steam used by the auxiliaries may be ascertained by allowing the engine alone to exhaust into the condenser, measuring the feed water supplied to the boiler and the water discharged by the air pump, and subtracting one from the other, after allowing for losses by leakage.

XII. *Coal Measurement.* — (a) *Commercial Tests:* In commercial tests of the combined engine and boiler equipment, or those made under ordinary conditions of commercial service, the test should, as pointed out in Article VII, extend over the entire period of the day; that is, twenty-four hours, or a number of days of that duration. Consequently, the coal consumption should be determined for the entire time. If the engine runs but a part of the time, and during the remaining portion the fires are banked, the measurement of coal should include that used for banking. It is well, however, in such cases, to determine separately the amount consumed during the time the engine is in operation and that consumed during the period while the fires are banked, so as to have complete data for purposes of analysis and comparison, using suitable precautions to obtain reliable measurements. The measurement of coal begins with the first firing, after cleaning the furnaces and burning down at the beginning of the test, as pointed out in Article VIII, and ends with the last firing, at the expiration of the allotted time.

(b) *Continuous Running Tests:* In continuous running tests which, as pointed out in Article VII, cover one or more periods which elapse between the cleaning of the fires, the same principle applies as that mentioned under the above heading (a); viz., the coal measurement

begins with the first firing, after cleaning and burning down, and the measurement ends with the last firing, before cleaning and burning down at the close of the trial.

(*c*) *Coal Tests in General:* When not otherwise specially understood, a coal test of a combined engine and boiler plant is held to refer to the commercial test above noted, and the measurement of coal should conform thereto.

In connection with coal measurements, whatever the class of tests, it is important to ascertain the percentage of moisture in the coal, the weight of ashes and refuse, and, where possible, the approximate and ultimate analysis of the coal, following all the methods and details advocated in the latest report of the Boiler Test Committee of the Society. (See Vol. XXI, p. 34.)

(*d*) *Other Fuels than Coal:* For all other solid fuels than coal the same directions in regard to measurement should be followed as those given for coal. If the boilers are run with oil or gas, the measurements relating to stopping and starting are much simplified, because the fuel is burned as fast as supplied, and there is no body of fuel constantly in the furnace, as in the case of using solid fuel. When oil is used, it should be weighed, and when gas is used, it should be measured in a calibrated gas meter or a gasometer.

XIII. *Indicated Horse Power.* — The indicated horse power should be determined from the average mean effective pressure of diagrams taken at intervals of twenty minutes, and at more frequent intervals if the nature of the test makes this necessary, for each end of each cylinder. With variable loads, such as those of engines driving generators for electric railroad work, and of rubber-grinding and rolling-mill engines, the diagrams cannot be taken too often. In cases like the latter, one method of obtaining suitable averages is to take a series of diagrams on the same blank card without unhooking the driving cord, and apply the pencil at successive intervals of ten seconds until two minutes' time or more has elapsed, thereby obtaining a dozen or more indications in the time covered. This tends to insure the determination of a fair average for that period. In taking diagrams for variable loads, as indeed for any load, the pencil should be applied long enough to cover several successive revolutions, so that the variations produced by the action of the governor may be properly recorded. To determine whether the governor is subject to what is called "racing" or "hunting," a "variation diagram" should be obtained; that is, one in which the pencil is applied a sufficient time to cover a complete cycle of variations. When the governor is found to be working in this manner, the defect should be remedied before proceeding with the test.

It is seldom necessary, as far as average power measurements are concerned, to obtain diagrams at precisely the same instant at the two ends of the cylinder, or at the same instant on all the cylinders, when there are more than one. All that is required is to take the diagrams at regular intervals. Should the diagrams vary so much among themselves that the average may not be a fair one, it signifies that they should be taken more frequently, and not that special care should be employed to obtain the diagrams of each set at precisely the same time. When diagrams are taken during the time when the engine is working up to speed at the start, or when a study of valve setting and steam distribution is being made, they should be taken at as nearly the same time as practicable. In cases where the diagrams are to be taken simultaneously, the best plan is to have an operator stationed at each indicator. This is desirable, even where an electric or other device is employed to operate all the instruments at once; for unless there are enough operators, it is necessary to open the indicator cocks some time before taking the diagrams and run the risk of clogging the pistons and heating the high-pressure springs above the ordinary working temperature.

The most satisfactory driving rig for indicating seems to be some form of well-made pantagraph, with driving cord of fine annealed wire leading to the indicator. The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal lengths when the same indicator is attached to either end of the cylinder, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

The use of a three-way cock and a single indicator connected to the two ends of the cylinder is not advised, except in cases where it is impracticable to use an indicator close to each end. If a three-way cock is used, the error produced should be determined and allowed for.

To determine the average power developed in cases where the engine starts from rest during the progress of the trial, as in a commercial test of a plant where the engine runs only a portion of the twenty-four hours, a number of diagrams should be taken during the period of getting up speed and applying the working load, the corresponding speed for each set of diagrams being counted. The power shown by these diagrams for the proportionate time should be included in the average for the whole run, and the duration should be the time the throttle valve is open.

XIV. *Testing Indicator Springs.* — To make a perfectly satisfactory comparison of indicator springs with standards, the calibration should be made, if this were practical, under the same conditions as those

pertaining to their ordinary use. Owing to the fact that the pressure of the steam in the indicator cylinder and the corresponding temperature are undergoing continual changes, it becomes almost impossible to compare the springs with any standard under such conditions. There must be a constant pressure during the time that the comparison is being made. Although the best that can be done is not altogether satisfactory, it seems that we must be content with it. To bring the conditions as nearly as possible to those of the working indicator, the steam should be admitted to the indicator as short a time as practicable for each of the pressures tried, and then the indicator cock should be closed and the steam exhausted therefrom before another pressure is tried. By this means the parts are heated and cooled somewhat the same as under the working conditions. We recommend, therefore, that for each required pressure the first step be to open and close the indicator cock a number of times in quick succession, then to quickly draw the line on the paper for the desired record, observing the gauge or other standard at the instant when the line is drawn. A corresponding atmospheric line is taken immediately after obtaining the line at the given pressure, so as to eliminate any difference in the temperature of the parts of the indicator. This appears to be a better method (although less readily carried on and requiring more care) than the one heretofore more commonly used, where the indicator cock is kept continually open and the pressure is gradually rising or falling through the range of comparison.

The calibration should be made for at least five points, two of these being for the pressure corresponding as near as may be to the initial and back pressures, and three for intermediate points equally distant.

For pressures above the atmosphere, the proper standard recommended is the dead-weight testing apparatus, or a reliable mercury column, or an accurate steam gauge proved correct, or of known error, by either of these standards. For pressures below the atmosphere the best standard to use is a mercury column.

The correct scale of spring to be used for working out the mean effective pressure of the diagrams should be the average based on the calibration, and this may be ascertained in the manner pointed out below.

XV. *Brake Horse Power.* — This term applies to the power delivered from the fly-wheel shaft of the engine. It is the power absorbed by a friction brake applied to the rim of the wheel or to the shaft. A form of brake is preferred that is self-adjusting to a certain extent, so that it will of itself tend to maintain a constant resistance at the rim of the wheel. One of the simplest brakes for comparatively small engines,

which may be made to embody this principle, consists of a cotton or hemp rope, or a number of ropes, encircling the wheel, arranged with weighing scales or other means for showing the strain. An ordinary band brake may also be constructed so as to embody the principle. The wheel should be provided with interior flanges for holding water used for keeping the rim cool.

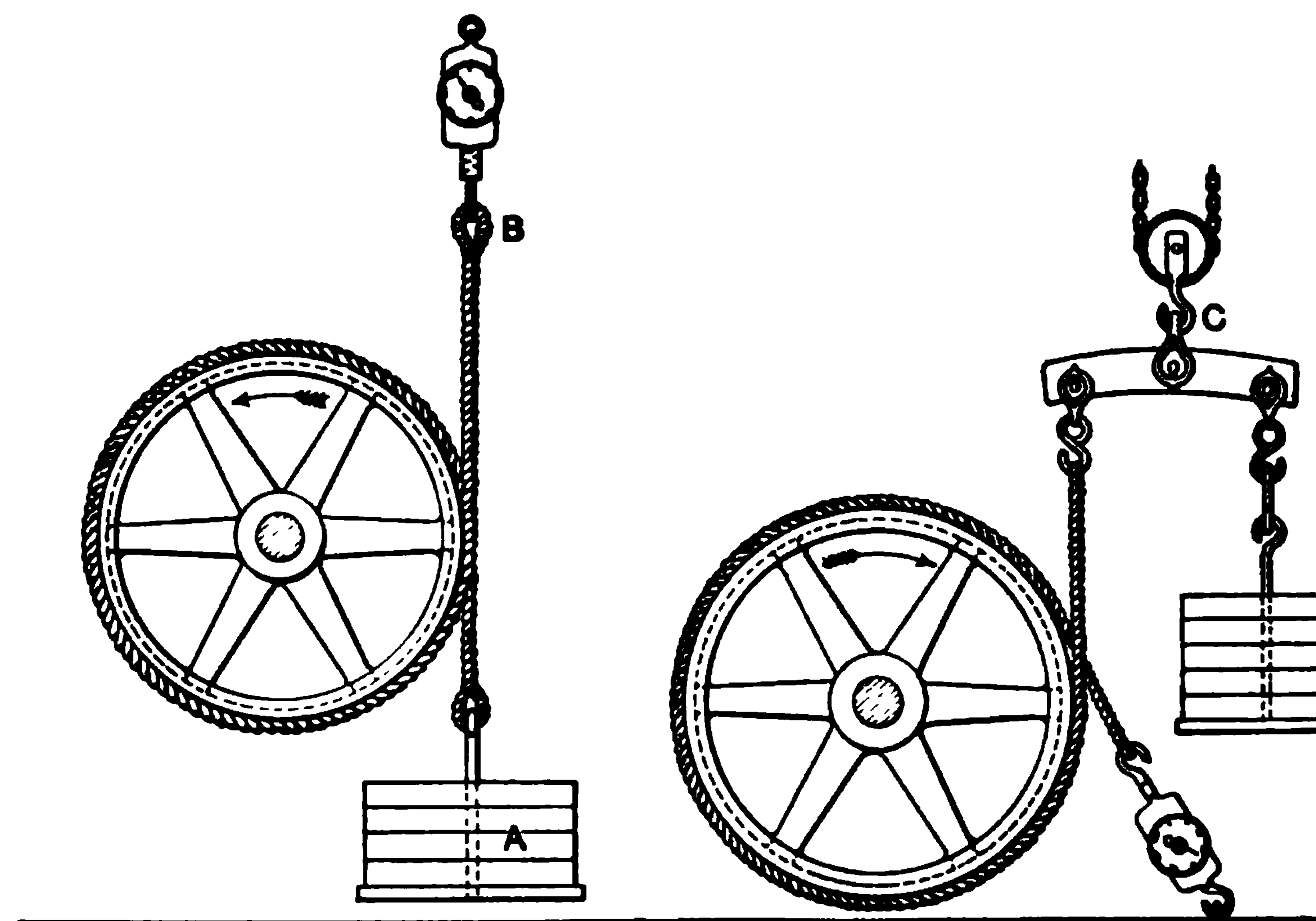


FIG. 454. FIG. 455.
Rope Brakes.

A self-adjusting rope brake is illustrated in Fig. 454, where it will be seen that, if the friction at the rim of the wheel increases, it will lift the weight A, which action will diminish the tension in the end B of the rope and thus prevent a further increase in the friction. The same device can be used for a band brake of the ordinary construction. Where space below the wheel is limited, a cross bar, C, supported by a chain tackle exactly at its center point may be used as shown in Fig. 455, thereby causing the action of the weight on the brake to be upward. A safety stop should be used with either form, to prevent the weights being accidentally raised more than a certain amount.

The water-friction brake is specially adapted for high speeds and has the advantage of being self-cooling. The Alden brake is also self-cooling and is capable of fine adjustment.

A water-friction brake is shown in Fig. 456. It consists of two circular disks, A and B, attached to the shaft C, and revolving in a case, E, between fixed planes. The space between the disks and planes is

supplied with running water, which enters at D and escapes at the cocks F, G, and H. The friction of the water against the surfaces constitutes a resistance which absorbs the desired power, and the heat generated within is carried away by the water itself. The water is thrown outward by centrifugal action and fills the outer portion of the case. The greater the depth of the ring of water the greater the

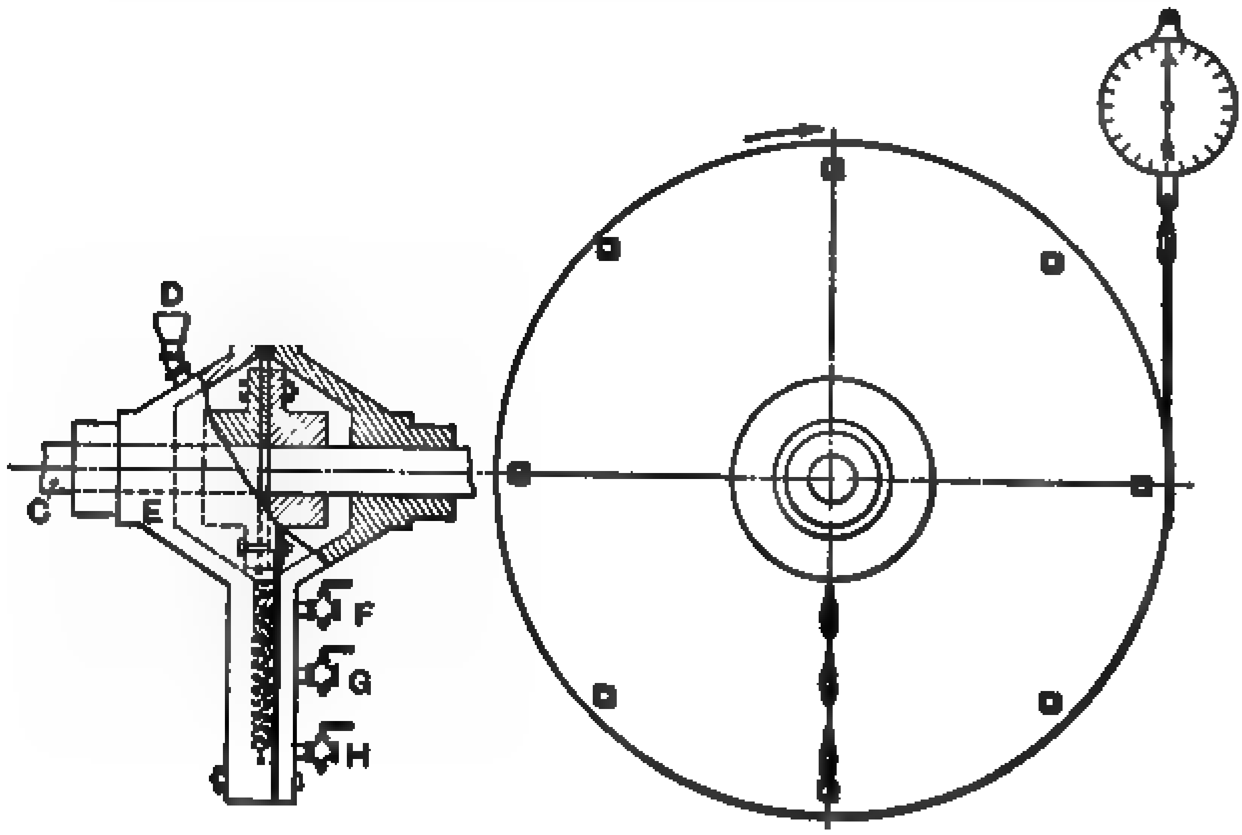


FIG. 456. Alden Absorption Dynamometer.

amount of power absorbed. By suitably adjusting the amount of water entering and leaving any desired power can be obtained. Water-friction brakes have been used successfully at speeds of over 20,000 revolutions per minute.

For description of the Alden brake, see *Transactions*, Vol. XI, p. 958.

XVI. *Quality of Steam.* — When ordinary saturated steam is used, its quality should be obtained by the use of a throttling calorimeter attached to the main steam pipe near the throttle valve. When the steam is superheated, the amount of superheating should be found by the use of a thermometer placed in a thermometer-well filled with mercury, inserted in the pipe. The sampling pipe for the calorimeter should, if possible, be attached to a section of the main pipe having a vertical direction, with the steam preferably passing upward, and the sampling nozzle should be made of a half-inch pipe, having at least 20 one-eighth-inch holes in its perforated surface. The readings of the calorimeter should be corrected for radiation of the instrument, or they should be referred to a normal reading, as pointed out below. If the steam is

superheated, the amount of superheating should be obtained by referring the reading of the thermometer to that of the same thermometer when the steam within the pipe is saturated, and not by taking the difference between the reading of the thermometer and the temperature of saturated steam at the observed pressure as given in a steam table.

XVII. *Speed.* — There are several reliable methods of ascertaining speed, or the number of revolutions of the engine crank-shaft per minute. The simplest is the familiar method of counting the number of turns for a period of one minute with the eye fixed on the second hand of a timepiece. Another is the use of a counter held for a minute or a number of minutes against the end of the main shaft. Another is the use of a reliable calibrated tachometer held likewise against the end of the shaft. The most reliable method, and the one we recommend, is the use of a continuous recording engine register or counter, taking the total reading each time that the general test data are recorded, and computing the revolutions per minute corresponding to the difference in the readings of the instrument. When the speed is above 250 revolutions per minute, it is almost impossible to make a satisfactory counting of the revolutions without the use of some form of mechanical counter.

The determination of variation of speed during a single revolution, or the effect of the fluctuation due to sudden changes of the load, is also desirable, especially in engines driving electric generators used for lighting purposes. There is at present no recognized standard method of making such determinations, and if such are desired, the method employed may be devised by the person making the test and described in detail in the report.

XVIII. *Recording the Data.* — Take note of every event connected with the progress of the trial whether it seems at the time to be important or unimportant. Record the time of every event and time of taking every weight and every observation. Observe the pressures, temperatures, water heights, speeds, etc., every twenty or thirty minutes when the conditions are practically uniform, and at much more frequent intervals if the conditions vary. Observations which concern the feed-water measurements should be made with special care at the expiration of each hour of the trial, so as to divide the tests into hourly periods and show the uniformity of the conditions and results as the test goes forward. Where the water discharged from a surface condenser is weighed it may be advisable to divide the test by this means into periods of less than one hour.

The data and observations of the test should be kept on properly prepared blanks or in notebooks containing columns suitably arranged for a clear record. As different observers have their own individual ideas as to how such records should be kept, no special form of log sheet is given as a necessary part of the code.

XIX. *Uniformity of Conditions.* — In a test having for an object the determination of the maximum economy obtainable from an engine, or where it is desired to ascertain with special accuracy the effect of predetermined conditions of operation, it is important that all the conditions under which the engine is operated should be maintained uniformly constant. This requirement applies especially to the pressure, the speed, the load, the rate of feeding the various supplies of water, the height of water in the gauge glasses, and the depth of water in the feed-water reservoir.

XX. *Analysis of Indicator Diagrams.* — (*a*) *Steam accounted for by the Indicator:* The simplest method of computing the steam accounted for by the indicator is the use of the formula

$$M = \frac{13750}{\text{M.E.P.}}[(C + E) \times Wc - (H + E) \times Wh],$$

which gives the weight in pounds per indicated horse power per hour. In this formula the symbol "M.E.P." refers to the mean effective pressure. In multiple-expansion engines this is the combined mean effective pressure referred to the cylinder in question. The symbol C refers to the proportion of the stroke completed at points on the expansion line of the diagram near the actual cut-off or release; the symbol H to the proportion of compression; and the symbol E to the proportion of clearance; all of which are determined from the indicator diagram. The symbol Wc refers to the weight of one cubic foot of steam at the cut-off or release pressure; and the symbol Wh to the weight of one cubic foot of steam at the compression pressure; these weights being taken from steam tables of recognized accuracy. The points near the cut-off and release on the expansion line and the point on the compression line are located as shown on the sample diagram, Fig. 457. They are the points in the case of the expansion and compression lines of the diagram which mark the complete closure of the valve. The point near the cut-off, for example, lies where the curve of expansion begins after the rounding of the diagram due to the wire-drawing which occurs while the valve is closing. This cut-off may be located by finding the point where the curve is tangent to a hyperbolic curve.

Should the point in the compression curve be at the same height as the point in the expansion curve, then $Wc = Wh$, and the formula becomes

$$\frac{13750}{\text{M.E.P.}} \times (C - H) \times Wc,$$

in which $(C - H)$ represents the distance between the two points divided by the length of the diagram.

When the load and all other conditions are substantially uniform, it is unnecessary to work up the steam accounted for by the indicator

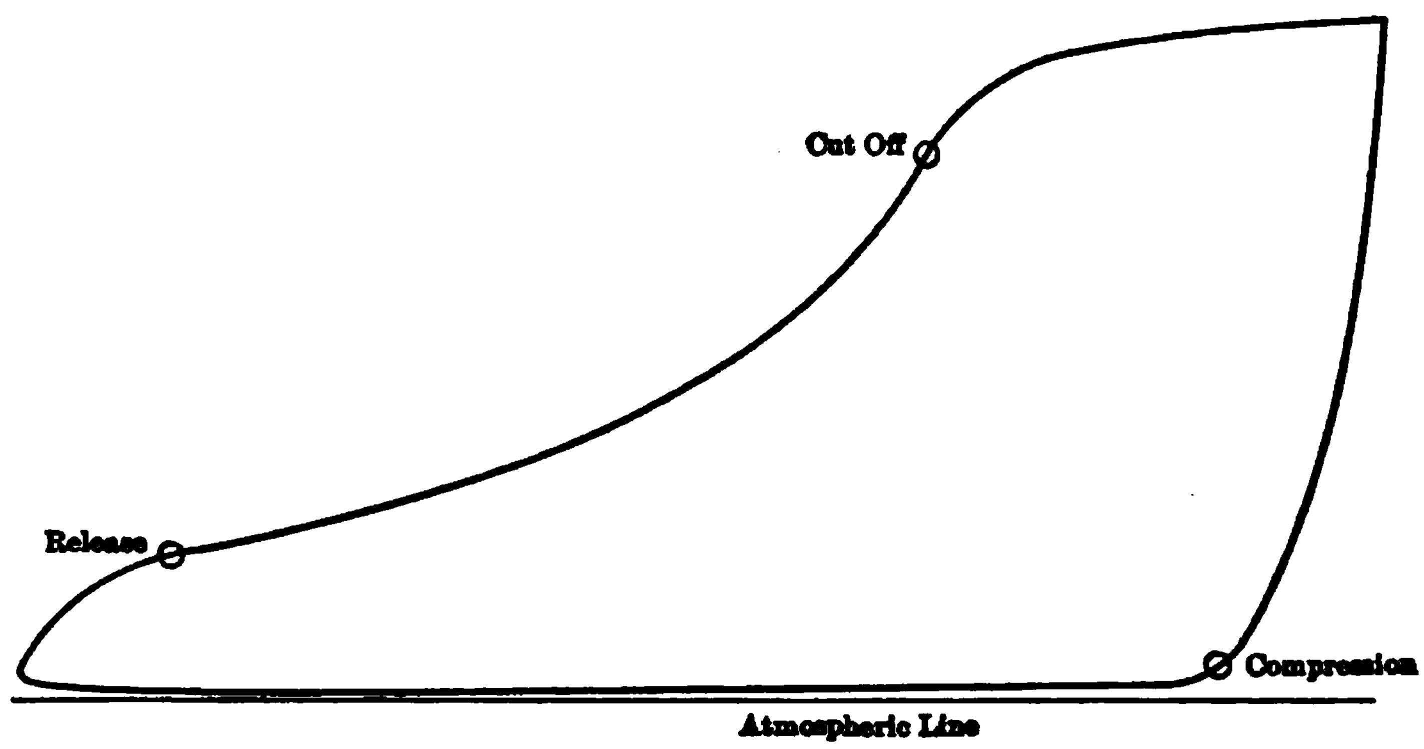


FIG. 457. Showing Points where "Steam Accounted for by Indicator" is Computed.

from all the diagrams taken. Five or more sample diagrams may be selected and the computations based on the samples instead of on the whole.

(b) *Sample Indicator Diagrams:* In order that the report of a test may afford complete information regarding the conditions of the test, sample indicator diagrams should be selected from those taken and copies appended to the tables of results. In cases where the engine is of the multiple-expansion type these sample diagrams may also be arranged in the form of a "combined" diagram.

(c) *The Point of Cut-off:* The term "cut-off" as applied to steam engines, although somewhat indefinite, is usually considered to be at an earlier point in the stroke than the beginning of the real expansion line. That the cut-off point may be defined in exact terms for commercial purposes, as used in steam-engine specifications and contracts, the Committee recommends that, unless otherwise specified, the *commercial cut-off*, which seems to be an appropriate expression for this term, be ascertained as follows: Through a point showing the maximum pressure

during admission draw a line parallel to the atmospheric line. Through the point on the expansion line near the actual cut-off, referred to in Section XX (*a*), draw a hyperbolic curve. The point where these two lines intersect is to be considered the *commercial cut-off* point. The percentage is then found by dividing the length of the diagram measured to this point by the total length of the diagram and multiplying the result by 100.

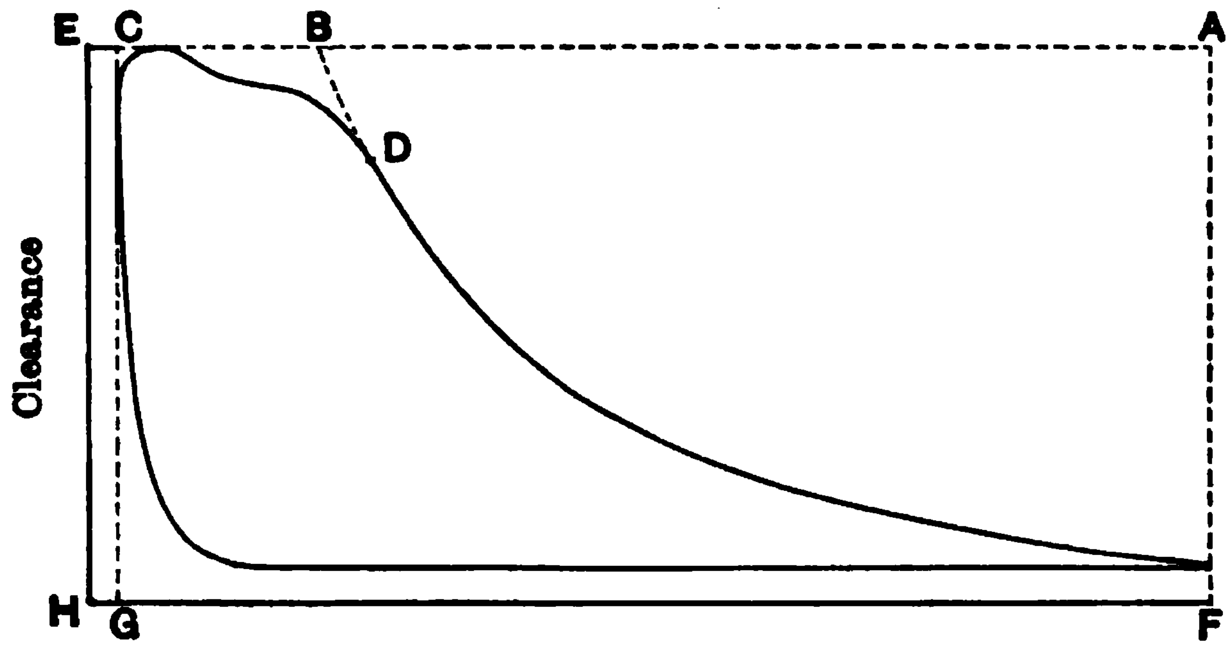


FIG. 458. Four Valve Engine, Slow Speed, Commercial Cut-off $= \frac{BC}{AC}$.

The principle involved in locating the commercial cut-off is shown in Figs. 458 and 459, the first of which represents a diagram from a slow-speed Corliss engine and the second a diagram from a single-valve high-speed engine. In the latter case where, owing to the fling of the

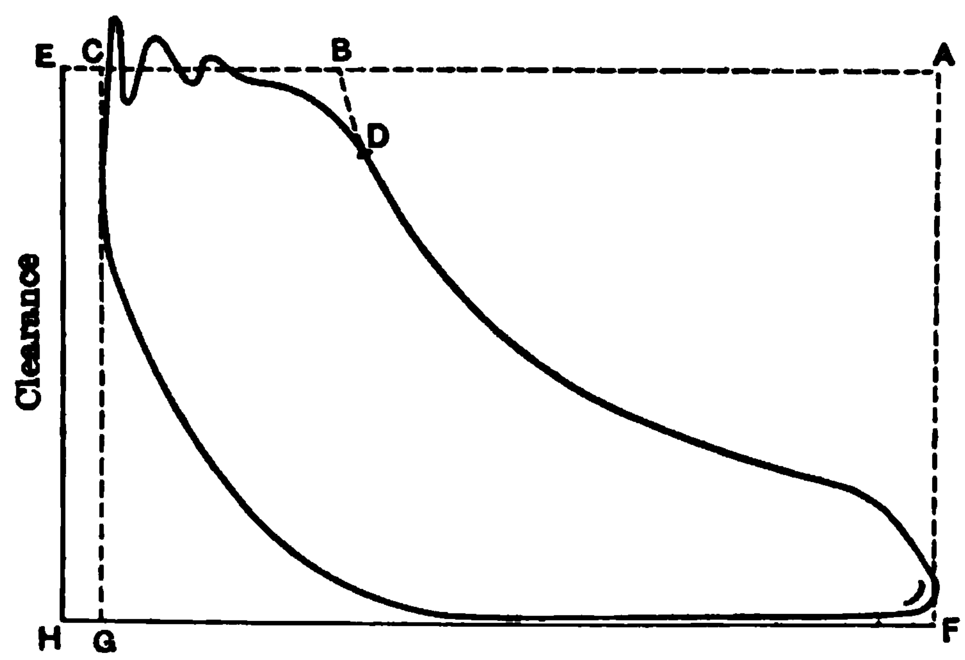


FIG. 459. Single Valve Engine, High-Speed, Commercial Cut-off $= \frac{BC}{AC}$.

pencil, the steam line vibrates, the maximum pressure is found by taking a mean of the vibrations at the highest point.

The *commercial cut-off* as thus determined is situated at an earlier

point of the stroke than the actual cut-off referred to in computing the "steam accounted for" by the indicator in Section XX (*a*).

(*d*) *Ratio of Expansion:* The ratio of expansion for a simple engine is determined by dividing the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple-expansion engine it is determined by dividing the net volume of the steam indicated by the low-pressure diagram at the end of the expansion line, assumed to be continued to the end of the stroke, by the net volume of the steam at the maximum pressure during admission to the high-pressure cylinder.

(*e*) *Diagram Factor:* The diagram factor is the proportion borne by the actual mean effective pressure measured from the indicator diagram to that of a diagram in which the various operations of admission, expansion, release, and compression are carried on under assumed conditions. The factor recommended refers to an ideal diagram which represents the maximum power obtainable from the steam accounted for by the indicator diagrams at the point of cut-off, assuming first that the engine has no clearance; second, that there are no losses through wire-drawing the steam either during the admission or the release; third, that the expansion line is a hyperbolic curve; and fourth, that the initial pressure is that of the boiler and the back pressure that of the atmosphere for a non-condensing engine and of the condenser for a condensing engine.

The diagram factor is useful for comparing the steam distribution losses in different engines, and is of special use to the engine designer, for by multiplying the mean effective pressure obtained from the assumed theoretical diagrams by it he will obtain the actual mean effective pressure that should be developed in an engine of the type considered. The expansion and compression curves are taken as hyperbolas, because such curves are ordinarily used by engine builders in their work, and a diagram based on such curves will be more useful to them than one where the curves are constructed according to a more exact law.

In cases where there is a considerable loss of pressure between the boiler and the engine, as where steam is transmitted from a central plant to a number of consumers, the pressure of the steam in the supply main should be used in place of the boiler pressure in constructing the diagrams.

XXI. *Standards of Economy and Efficiency.* — The hourly consumption of heat, determined by employing the actual temperature of the feed water to the boiler, as pointed out in Article IX of the Code, divided

by the indicated and brake horse power, that is, the number of heat units consumed per indicated and per brake horse power per hour, is the standard of engine efficiency recommended by the Committee. The consumption *per hour* is chosen rather than the consumption per minute, so as to conform with the designation of time applied to the more familiar units of coal and water measurement which have heretofore been used. The British standard, where the temperature of the feed water is taken as that corresponding to the temperature of the back-pressure steam, allowance being made for any drips from jackets or reheaters, is also included in the tables.

It is useful in this connection to express the efficiency in its more scientific form, or what is called the "thermal efficiency ratio." The thermal efficiency ratio is the proportion which the heat equivalent of the power developed bears to the total amount of heat actually consumed, as determined by test. The heat converted into work represented by one horse power is 1,980,000 foot-pounds per hour, and this divided by 778 equals 2545 British thermal units. Consequently the thermal efficiency ratio is expressed by the fraction

$$\frac{2545}{\text{B.T.U. per H.P. per hour}}.$$

XXII. *Heat Analysis.* — For certain scientific investigations it is useful to make a heat analysis of the diagram to show the interchange of heat from steam to cylinder walls, etc., which is going on within the cylinder. This is unnecessary for commercial tests.

XXIII. *Temperature-Entropy Diagram.* — The study of the heat analysis is facilitated by the use of the temperature-entropy diagram in which areas represent quantities of heat, the coördinates being the absolute temperature and entropy. Such a diagram is shown in Fig. 460.

When the quantities given in the steam tables are plotted, two curves, AA and BB, are obtained which may be termed the water line and the steam line, AA being the logarithmic curve if the specific heat of the water is taken as constant. The diagram refers to a unit weight of the agent, and the heat necessary to raise a pound of water from the temperature ma to the temperature pa' and evaporate it at that temperature is represented by the area $aa'b'qm$. If the steam be now expanded adiabatically the temperature will fall to qs and x per cent $= \frac{as}{ab}$ will remain as steam, the rest being liquefied. If the steam is now rejected, it carries away with it the heat $sqma$, the work

area being $a'b'sa$, from which must be deducted the work w (expressed in heat units) to pump a pound of water into the boiler. The efficiency of this cycle is evidently

$$\frac{h + L_1 - xL_2 - w}{h + L_1},$$

in which

$$x = \frac{ar + a'b'}{ab} = \frac{\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1}}{\frac{L_2}{T_2}}.$$

By the action of the walls a portion of the steam is liquefied prior to the expansion, which therefore begins at e, and since the cooling

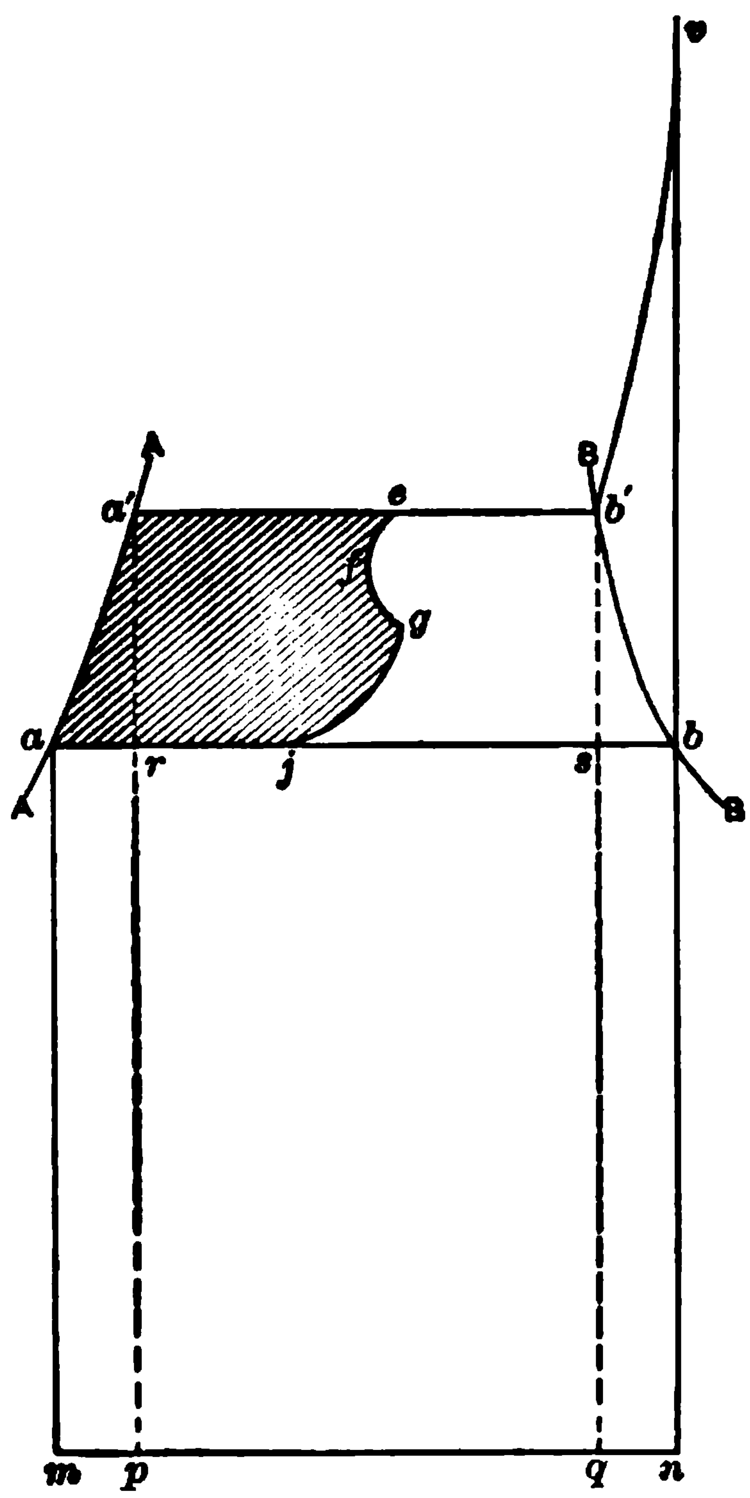


Fig. 460.

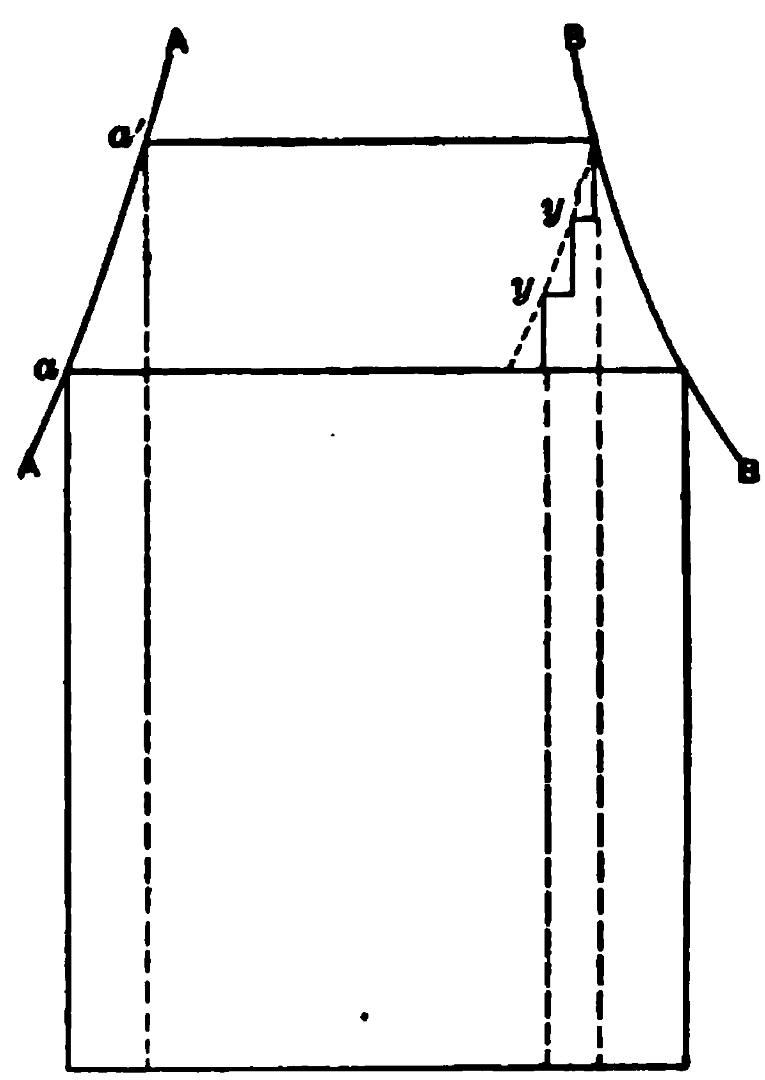


Fig. 461.

Temperature-Entropy Diagrams.

action of the walls continues, the expansion line falls off to ef, from which point a reverse action takes place and the expansion line bends

over to g. Finally, since the release takes place before the condenser temperature is reached, the heat rejection starts at g, following a line of equal volume until the exhaust port temperature is reached at j. If heat is added during expansion enough to keep the steam theoretically saturated, as, for example, by a water jacket, such additional heat is represented by the area $b'bnq$, and the additional work obtained by the triangle $b'bs$. If the steam is superheated sufficiently to give by expansion theoretically dry steam at the end, such additional heat is represented by the area $b'vnq$ and the additional work by $b'vbs$. Neither of these extra amounts of work is realized in practice, and it is evident from the diagram that the heat thus applied is in both cases less efficient than in the principal cycle. Nevertheless the action in each case is to bring the point e nearer the point b' and to effect a notable net economy.

The Carnot cycle would be obtained if in the Rankine cycle the rejection of heat were stopped at r and the temperature of the mixture raised to a' by compression. This cannot be practically accomplished, but a system of feed-water heaters has been suggested and exemplified in the Nordberg engine, which is theoretically a close equivalent to it. Where steam is expanded in say three cylinders, the feed water may be successively heated from the receiver intermediate between each pair, the effect of which is illustrated in Fig. 461. The expansion line follows the heavy line, being carried over to y by the first feed-water heater and to y' by the second feed-water heater. With an infinite number of such feed-water heaters, the line yy' would be parallel to aa', and the cycle equivalent to that of Carnot.

XXIV. *Ratio of Economy of an Engine to that of an Ideal Engine.* — The ideal engine recommended for obtaining this ratio is that which was adopted by the Committee appointed by the Civil Engineers of London to consider and report a standard thermal efficiency for steam engines. This engine is one which follows the Rankine cycle, where steam at a constant pressure is admitted into the cylinder with no clearance, and after the point of cut-off is expanded adiabatically to the back pressure. In obtaining the economy of this engine the feed water is assumed to be returned to the boiler at the exhaust temperature. Such a cycle is preferable to the Carnot for the purpose at hand, because the Carnot cycle is theoretically impossible for an engine using superheated steam produced at a constant pressure, and the gain in efficiency for superheated steam corresponding to the Carnot efficiency will be much greater than that possible for the actual cycle.

The ratio of the economy of an engine to that of the ideal engine is obtained by dividing the heat consumption per indicated

horse power per minute for the ideal engine by that of the actual engine.

XXV. *Miscellaneous.* — In the case of tests of combined engines and boiler plants, where the full data of the boiler performance is to be determined, reference should be made to the directions given by the Boiler Test Committee of the Society, Code of 1899. (See Vol. XXI, p. 34.)

In tests made for scientific research, and in those made on special forms of engines, the line of procedure must be varied according to the special objects in view, and it has been deemed unnecessary to go into particulars applying to such tests.

In testing steam pumping engines and locomotives in accordance with the standard methods of conducting such tests, recommended by the committees of the Society, reference should be made to the reports of those committees in the *Transactions*, Vol. XII, p. 530, and in Vol. XIV, p. 1312.

XXVI. *Report of Test.* — The data and results of the test should be reported in the manner and in the order outlined in one of the following tables, the first of which gives, it is hoped, a complete summary of all the data and results as applied not only to the standard heat-unit test but also to tests of combined engine and boiler for determining all questions of performance, whatever the class of service; the second refers to a short form of report giving the necessary data and results for the standard heat test; and the third to a short form of report for a feed-water test. It is the intention that the tables should be full enough to apply to any type of engine, but where not so, or where special data and results are determined, additional results may be inserted under the appropriate headings. Although these forms are arranged so as to be used for expressing the principal data and results of tests of pumping engines and locomotives, as well as for all other classes of steam engines, it is not the intention that they shall supplant the forms recommended by the committees on Duty Trials and Locomotives in cases where the full report of a test of such engines is desired.

It is recommended that any report be supplemented by a chart in which the data of the test are graphically presented. (As an example of such a chart as applied to a boiler test, see Vol. XXI, p. 104.)

TABLE NO. 1.

Not reprinted here. See Trans. A.S.M.E. 24–702.

TABLE NO. 2.

DATA AND RESULTS OF STANDARD HEAT TEST OF STEAM ENGINE.

Arranged according to the Short Form advised by the Engine Test Committee of the American Society of Mechanical Engineers. Code of 1902.

1. Made by .... of ....
   on engine located at ....
   to determine ....
   ....
2. Date of trial ....
3. Type and class of engine; also of condenser ....
   ....

		1st Cyl.	2d Cyl.	3d Cyl.
4. Dimensions of main engine				
(a) Diameter of cylinder	in.			
(b) Stroke of piston	ft.			
(c) Diameter of piston rod	in.			
(d) Average clearance	p. c.			
(e) Ratio of volume of cylinder to high-pressure cylinder				
(f) Horse-power constant for one pound mean effective pressure and one revolution per minute				

5. Dimensions and type of auxiliaries ....
   ....

Total Quantities, Time, etc.

6. Duration of test	hours.
7. Total water fed to boilers from main source of supply	pounds.
8. Total water fed from auxiliary supplies:	
(a)	"
(b)	"
(c)	"
9. Total water fed to boilers from all sources	pounds.
10. Moisture in steam or superheating near throttle	p. c. or deg.
11. Factor of correction for quality of steam	
12. Total dry steam consumed for all purposes	pounds

Hourly Quantities.

13. Water fed from main source of supply	"
14. Water fed from auxiliary supplies:	
(a)	"
(b)	"
(c)	"
15. Total water fed to boilers per hour	"
16. Total dry steam consumed per hour	"
17. Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant	"
18. Net dry steam consumed per hour by engine and auxiliaries	"

Pressures and Temperatures (Corrected).

19. Pressure in steam pipe near throttle by gauge.................. lb. per sq. in.
20. Barometric pressure of atmosphere in inches of mercury......... inches.
21. Pressure in receivers by gauge.............................. lb. per sq. in.
22. Vacuum in condenser in inches of mercury..................... inches.
23. Pressure in jackets and reheaters by gauge..................... lb. per sq. in.
24. Temperature of main supply of feed water..................... degrees F.
25. Temperature of auxiliary supplies of feed water:
    (a).. "
    (b).. "
    (c).. "
26. Ideal feed-water temperature corresponding to pressure of steam in the exhaust pipe, allowance being made for heat derived from jacket or reheater drips.......................... "

Data Relating to Heat Measurement.

27. Heat units per pound of feed water, main supply............... B.T.U.
28. Heat units per pound of feed water, auxiliary supplies:
    (a).. "
    (b).. "
    (c).. "
29. Heat units consumed per hour, main supply................... "
30. Heat units consumed per hour, auxiliary supplies:
    (a).. "
    (b).. "
    (c).. "
31. Total heat units consumed per hour for all purposes............ "
32. Loss of heat per hour due to leakage of plant, drips, etc......... "
33. Net heat units consumed per hour:
    (a) By engine alone.................................. "
    (b) By auxiliaries................................... "
34. Heat units consumed per hour by engine alone, reckoned from temperature given in line 26.......................... "

Indicator Diagrams.

	1st Cyl.	2d Cyl.	3d Cyl.
35. Commercial cut-off in per cent of stroke........			
36. Initial pressure in pounds per square inch above atmosphere..........................			
37. Back pressure at mid-stroke above or below atmosphere in pounds per square inch.......			
38. Mean effective pressure in pounds per square inch................................			
39. Equivalent mean effective pressure in pounds per square inch:			
(a) Referred to first cylinder................			
(b) Referred to second cylinder.............			
(c) Referred to third cylinder...............			

	1st Cyl.	2d Cyl.	3d Cyl.
40. Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves..............			
Pressure above zero in pounds per square inch:			
(*a*) Near cut-off..........................			
(*b*) Near release..........................			
(*c*) Near beginning of compression...........			
Percentage of stroke at points where pressures are measured:			
(*a*) Near cut-off..........................			
(*b*) Near release..........................			
(*c*) Near beginning of compression...........			
41. Steam accounted for by indicator in pounds per I.H.P. per hour:			
(*a*) Near cut-off..........................			
(*b*) Near release..........................			
42. Ratio of expansion..........................			

Speed

43. Revolutions per minute.................................... revolutions.

Power.

44. Indicated horse power developed by main-engine cylinders:
First cylinder.. horse power.
Second cylinder.. "
Third cylinder... "
Total... "
45. Brake horse power developed by engine...................... "

*Standard Efficiency and other Results.**

46. Heat units consumed by engine and auxiliaries per hour:
(*a*) per indicated horse power............................ B.T.U.
(*b*) per brake horse power................................ "
47. Equivalent standard coal in pounds per hour:
(*a*) per indicated horse power............................ pounds.
(*b*) per brake horse power................................ "
48. Heat units consumed by main engine per hour corresponding to ideal maximum temperature of feed water given in line 26, British standard:
(*a*) per indicated horse power............................ B.T.U.
(*b*) per brake horse power................................ "
49. Dry steam consumed per indicated horse power per hour:
(*a*) Main cylinders, including jackets .... pounds.
(*b*) Auxiliary cylinders................................... "
(*c*) Engine and auxiliaries................................. "

* The horse power referred to above (items 46–50) is that of the main engine, exclusive of auxiliaries.

50. Dry steam consumed per brake horse power per hour:
    (a) Main cylinders, including jackets .... pounds.
    (b) Auxiliary cylinders .... "
    (c) Engine and auxiliaries .... "
51. Percentage of steam used by main-engine cylinders accounted for by indicator diagrams, near cut-off of high-pressure cylinder per cent.

Additional Data.

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is used. Also give copies of indicator diagrams nearest the mean, and the corresponding scales.

TABLE NO. 3.

DATA AND RESULTS OF FEED-WATER TEST OF STEAM ENGINE.

Arranged according to the Short Form advised by the Engine Test Committee of the American Society of Mechanical Engineers. Code of 1902.

1. Made by .... of ....
   on engine located at ....
   to determine ....
   ....
2. Date of trial ....
3. Type of engine (simple, compound, or other multiple expansion; condensing or non-condensing) ....
   ....
4. Class of engine (mill, marine, locomotive, pumping, electric, or other) ....
5. Rated power of engine ....
6. Name of builders ....
7. Number and arrangement of cylinders of engine; how lagged; type of valves and of condensers ....

	1st Cyl.	2d Cyl.	3d Cyl.
8. Dimensions of engine			
(a) Single or double acting			
(b) Cylinder dimensions:			
Bore .... in.			
Stroke .... ft.			
Diameter of piston rod .... in.			
Diameter of tail rod .... in.			
(c) Clearance in per cent of volume displaced by piston per stroke:			
Head end			
Crank end			
Average			
(d) Ratio of volume of each cylinder to volume of high-pressure cylinder....			
(e) Horse-power constant for one pound mean effective pressure and one revolution per minute			

Total Quantities, Time, etc.

9. Duration of test........	hours.
10. Water fed to boilers from main source of supply........	pounds.
11. Water fed from auxiliary supplies:	
(a)........	"
(b)........	"
(c)........	"
12. Total water fed from all sources........	"
13. Moisture in steam or superheating near throttle *........	p. c. or deg.
14. Factor of correction for quality of steam........	
15. Total dry steam consumed for all purposes........	pounds.

Hourly Quantities.

16. Water fed from main source of supply........	"
17. Water fed from auxiliary supplies:	
(a)........	"
(b)........	"
(c)........	"
18. Total water fed to boilers per hour........	"
19. Total dry steam consumed per hour........	"
20. Loss of steam and water per hour due to leakage of plant, drips, etc........	"
21. Net dry steam consumed per hour by engine and auxiliaries........	"
22. Dry steam consumed per hour:	
(a) Main cylinders........	"
(b) Jackets and reheaters........	"

Pressures and Temperatures (Corrected).

23. Steam pipe pressure near throttle, by gauge........	lb. per sq. in.
24. Barometric pressure of atmosphere in inches of mercury........	inches.
25. Pressure in first receiver by gauge........	lb. per sq. in.
26. Pressure in second receiver by gauge........	"
27. Vacuum in condenser:	
(a) In inches of mercury........	inches.
(b) Corresponding total pressure........	lb. per sq. in.
28. Pressure in steam jackets by gauge........	lb. per sq. in.
29. Pressure in reheater by gauge........	"
30. Superheating of steam in first receiver........	degrees F.
31. Superheating of steam in second receiver........	"

Indicator Diagrams.

	1st Cyl.	2d Cyl.	3d Cyl.
32. Commercial cut-off in per cent of stroke........			
33. Initial pressure in pounds per square inch above atmosphere........			

* In case of superheated steam engines, determine, if practicable, the temperature of the steam in each cylinder.

	1st Cyl.	2d Cyl.	3d Cyl.
34. Back pressure at mid-stroke above or below atmosphere in pounds per square inch			
35. Mean effective pressure in pounds per square inch			
36. Equivalent mean effective pressure in pounds per square inch per indicated horse power......			
(a) Referred to first cylinder.			
(b) Referred to second cylinder.			
(c) Referred to third cylinder.			
37. Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves..............			
Pressures above zero in pounds per square inch:			
(a) Near cut-off..........................			
(b) Near release..........................			
(c) Near beginning of compression...........			
Percentage of stroke at points where pressures are measured:			
(a) Near cut-off..........................			
(b) Near release..........................			
(c) Near beginning of compression...........			
38. Aggregate M.E.P. in pounds per square inch referred to each cylinder given in heading			
39. Mean back pressure above zero, pounds per square inch............................			
40. Steam accounted for in pounds per indicated horse power per hour:			
(a) Near cut-off..........................			
(b) Near release..........................			

41. Ratio of expansion:

(a) Commercial..

(b) Ideal..

Speed.

42. Revolutions per minute.................................... revolutions.

43. Piston speed per minute.................................... feet.

Power.

44. Indicated horse power developed by main-engine cylinders:

First cylinder.................................... horse power.

Second cylinder.................................... "

Third cylinder.................................... "

Total.................................... "

Efficiency Results.

45. Dry steam consumed per indicated horse power per hour:

(a) Main cylinder, including jackets........................	pounds.
(b) Auxiliary cylinders, etc..................................	"
(c) Engine and auxiliaries....................................	"

46. Percentage of steam used by main-engine cylinders accounted for by indicator diagrams:

	1st Cyl.	2d Cyl.	3d Cyl.
(a) Near cut-off..........................			
(b) Near release..........................			

Sample Diagrams.

Copies of indicator diagrams, nearest the mean, with corresponding scales, should be given in connection with table.

APPENDIX D.

PROPERTIES OF SATURATED STEAM.* (Marks and Davis.)

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
P	t	q	r	$\lambda = r + q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
† 0.1	35.00	3.02	1071.7	1074.7	1017.3	54.0	0.0062	2.1666	2.1728	2938.0	0.000430
† 0.2	53.15	21.09	1061.6	1082.7	1005.2	56.2	0.0423	2.0704	2.1127	1540.0	0.000679
† 0.3	64.49	32.57	1055.3	1087.9	997.7	57.6	0.0640	2.0135	2.0775	1041.0	0.000961
0.4	72.91	40.95	1050.6	1091.6	997.7	58.5	0.0800	1.9730	2.0530	794.0	0.001259
0.5	79.68	47.64	1047.0	1094.6	987.6	59.3	0.0923	1.9415	2.0338	643.0	0.001555
0.6	85.32	53.34	1043.8	1097.1	983.9	59.9	0.1029	1.9155	2.0184	541.0	0.001850
0.7	90.18	58.18	1041.1	1099.3	980.7	60.4	0.1117	1.8936	2.0053	467.0	0.002143
0.8	94.46	62.45	1038.7	1101.2	977.8	61.0	0.1195	1.8747	1.9942	412.0	0.002431
0.9	98.33	66.31	1036.6	1102.9	975.2	61.4	0.1265	1.8578	1.9843	369.7	0.002719
1	101.83	69.8	1034.6	1104.4	972.9	61.7	0.1327	1.8427	1.9754	333.0	0.00300
2	126.15	94.0	1021.0	1115.0	956.7	64.3	0.1749	1.7431	1.9180	173.5	0.00576
3	141.52	109.4	1012.3	1121.6	946.4	65.8	0.2008	1.6840	1.8848	118.5	0.00845
4	153.01	120.9	1005.7	1126.5	938.6	67.0	0.2198	1.6416	1.8614	90.5	0.01107
5	162.28	130.1	1000.3	1130.5	932.4	68.0	0.2348	1.6084	1.8432	73.33	0.01364
6	170.06	137.9	995.8	1133.7	927.0	68.8	0.2571	1.5814	1.8285	71.89	0.01616
7	176.85	144.7	991.8	1136.5	922.4	69.4	0.2579	1.5582	1.8161	53.56	0.01867
8	182.86	150.8	988.2	1139.0	918.2	70.0	0.2673	1.5380	1.8053	47.27	0.02115
9	188.27	156.2	985.0	1141.1	914.4	70.6	0.2756	1.5202	1.7958	42.36	0.02361
10	193.22	161.1	982.0	1143.1	910.9	71.1	0.2832	1.5042	1.7874	38.38	0.02606
11	197.75	165.7	979.2	1144.9	907.8	71.5	0.2902	1.4895	1.7797	35.10	0.02849
12	201.96	169.9	976.6	1146.5	904.8	71.8	0.2967	1.4760	1.7727	32.36	0.03090

* Courtesy of the Publishers, Longmans, Green & Co. † Interpolated.

13	205.87	173.8	974.2	1148.0	902.0	72.2	0.3025	1.4639	1.7664	30.03	0.03330
14	209.55	177.5	971.9	1149.4	899.3	72.6	0.3081	1.4523	1.7604	28.02	0.03569
14.7	212.00	180.0	970.4	1150.4	897.6	72.9	0.3118	1.4447	1.7565	26.79	0.03732
15	213.0	181.0	969.7	1150.7	896.8	72.9	0.3133	1.4416	1.7549	26.27	0.03806
20	228.0	196.1	960.0	1156.2	885.8	74.3	0.3355	1.3965	1.7320	20.08	0.04980
25	240.1	208.4	952.0	1160.4	876.8	75.3	0.3532	1.3604	1.7136	16.30	0.0614
30	250.3	218.8	945.1	1163.9	869.0	76.2	0.3680	1.3311	1.6991	13.74	0.0728
35	259.3	227.9	938.9	1166.8	862.1	76.9	0.3868	1.3060	1.6868	11.89	0.0841
40	267.3	236.1	933.3	1169.4	855.9	77.6	0.3920	1.2841	1.6761	10.49	0.0953
45	274.5	243.4	928.2	1171.6	850.3	78.1	0.4021	1.2644	1.6665	9.39	0.1065
50	281.0	250.1	923.5	1173.6	845.0	78.6	0.4113	1.2468	1.6581	8.51	0.1175
55	287.1	256.3	919.0	1175.4	840.2	78.9	0.4196	1.2309	1.6505	7.78	0.1285
60	292.7	262.1	914.9	1177.0	835.6	79.7	0.4272	1.2160	1.6432	7.17	0.1394
65	298.0	265.7	911.0	1178.5	831.4	79.8	0.4344	1.2034	1.6368	6.65	0.1503
70	302.9	272.6	907.2	1179.8	827.3	80.1	0.4411	1.1896	1.6307	6.20	0.1612
75	307.6	277.4	903.7	1181.8	823.5	80.5	0.4474	1.1778	1.6252	5.81	0.1721
80	312.0	282.0	900.3	1182.3	819.8	80.7	0.4535	1.1665	1.6200	5.47	0.1829
85	316.3	286.3	897.1	1183.4	816.3	81.0	0.4590	1.1561	1.6151	5.16	0.1937
90	320.3	290.5	893.9	1184.4	813.0	81.2	0.4644	1.1461	1.6105	4.89	0.2044
95	324.0	294.5	890.9	1185.4	809.7	81.5	0.4694	1.1367	1.6061	4.65	0.2151
100	327.8	298.3	888.0	1186.3	806.6	81.7	0.4743	1.1277	1.6020	4.429	0.2258
105	331.4	302.0	885.2	1187.2	803.6	81.9	0.4789	1.1191	1.5980	4.230	0.2365
110	334.8	305.5	882.5	1188.0	800.7	82.1	0.4834	1.1108	1.5942	4.047	0.2472
115	338.1	309.0	879.8	1188.8	797.9	82.3	0.4877	1.1030	1.5907	3.880	0.2577
120	341.3	312.3	877.2	1189.6	795.2	82.5	0.4919	1.0954	1.5873	3.726	0.2683
125	344.4	315.5	874.7	1190.3	792.6	82.6	0.4959	1.0880	1.5839	3.583	0.2791
130	347.4	318.6	872.3	1191.0	790.0	82.8	0.4998	1.0809	1.5807	3.452	0.2897
135	350.3	321.7	869.9	1191.6	787.5	82.9	0.5035	1.0742	1.5777	3.331	0.3002
140	353.1	324.6	867.6	1192.2	785.0	83.0	0.5072	1.0675	1.5747	3.219	0.3107
145	355.8	327.4	865.4	1192.8	782.7	83.2	0.5107	1.0612	1.5719	3.112	0.3213

PROPERTIES OF SATURATED STEAM — (*Continued*).

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
p	t	q	r	$\lambda = r + q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
150	358.5	330.2	863.2	1193.4	780.4	83.3	0.5142	1.0550	1.5692	3.012	0.3320
155	361.0	332.9	861.0	1194.0	778.1	83.5	0.5175	1.0489	1.5664	2.920	0.3425
160	363.6	335.6	858.8	1194.5	775.8	83.6	0.5208	1.0431	1.5639	2.834	0.3529
165	366.0	338.2	856.8	1195.0	773.6	83.7	0.5239	1.0376	1.5615	2.753	0.3633
170	368.5	340.7	854.7	1195.4	771.5	83.8	0.5269	1.0321	1.5590	2.675	0.3738
175	370.8	343.2	852.7	1195.9	769.4	83.9	0.5299	1.0268	1.5567	2.602	0.3843
180	373.1	345.6	850.8	1196.4	767.4	84.0	0.5328	1.0215	1.5543	2.533	0.3948
185	375.4	348.0	848.8	1196.8	765.4	84.1	0.5356	1.0164	1.5520	2.468	0.4052
190	377.6	350.4	846.9	1197.3	763.4	84.2	0.5384	1.0114	1.5498	2.406	0.4157
195	379.8	352.7	845.0	1197.7	761.4	84.3	0.5410	1.0066	1.5476	2.346	0.4262
200	381.9	354.9	843.2	1198.1	759.5	84.4	0.5437	1.0019	1.5456	2.290	0.437
205	384.0	357.1	841.4	1198.5	757.6	84.5	0.5463	0.9973	1.5436	2.237	0.447
210	386.0	359.2	839.6	1198.8	755.8	84.5	0.5488	0.9928	1.5416	2.187	0.457
215	388.0	361.4	837.9	1199.2	754.0	84.6	0.5513	0.9885	1.5398	2.138	0.468
220	389.9	363.4	836.2	1199.6	752.3	84.7	0.5538	0.9841	1.5379	2.091	0.478
225	391.9	365.5	834.4	1199.9	750.5	84.7	0.5562	0.9799	1.5361	2.046	0.489
230	393.8	367.5	832.8	1200.2	748.8	84.8	0.5586	0.9758	1.5341	2.004	0.499
240	397.4	371.4	829.5	1200.9	745.4	85.0	0.5633	0.9676	1.5309	1.924	0.520
250	401.1	375.2	826.3	1201.5	742.0	85.1	0.5676	0.9600	1.5276	1.850	0.541
275	409.5	384.2	818.6	1202.8	734.2	85.3	0.5780	0.9419	1.5199	1.686	0.593
300	417.5	392.7	811.3	1204.1	726.8	85.6	0.5878	0.9251	1.5129	1.551	0.645

APPENDIX E.

EQUIVALENT VALUES OF ELECTRICAL AND MECHANICAL UNITS.

1 KILOWATT HOUR =

- 1,000 watt hours
- 1.34 horse-power hours
- 2,654,200 foot-pounds
- 3,600,000 joules
- 367,000 kilogram meters
- 3.53 pounds of water evaporated from and at 212° F.

1 HORSE-POWER HOUR =

- 0.746 kilowatt hours
- 1,980,000 foot-pounds
- 2,545 B.T.U.
- 273,740 kilogram meters
- 2.64 pounds of water evaporated from and at 212° F.

1 KILOWATT =

- 1,000 watts
- 1.34 horse power
- 1.358 cheval-vapeur
- 2,654,200 foot-pounds per hour
- 44,240 foot-pounds per minute
- 737.3 foot-pounds per second
- 3,412 B.T.U. per hour
- 56.9 B.T.U. per minute
- 0.948 B.T.U. per second
- 3.53 pounds of water evaporated from and at 212° F.

1 HORSE POWER =

- 746 watts
- 0.746 kilowatts
- 1.0136 cheval-vapeur
- 33,000 foot-pounds per minute
- 550 foot-pounds per second
- 2,545 B.T.U. per hour
- 42.4 B.T.U. per minute
- 0.707 B.T.U. per second
- 2.64 pounds of water evaporated from and at 212° F.

1 JOULE =

- 1 watt second
- 0.000000278 kilowatt hour
- 0.102 kilogram meter
- 0.0009477 B.T.U.
- 0.7373 foot-pound

1 FOOT-POUND =

- 1.356 joules
- 0.1383 kilogram meter
- 0.000000377 kilowatt hour
- 0.001285 B.T.U.
- 0.0000005 horse-power hour

1 WATT =

- 1 joule per second
- 0.00134 horse power
- 3.412 B.T.U. per hour
- 0.7373 foot-pound per second
- 0.0035 pound water evaporated from and at 212° F.

1 B.T.U. =

- 1,055 watt seconds
- 778 foot-pounds
- 107.6 kilogram meters
- 0.000293 kilowatt hour
- 0.000393 horse-power hour
- 0.001036 pound water evaporated from and at 212° F.

1 KILOGRAM-METER =

- 7.233 foot-pounds
- 0.00000365 horse-power hour
- 0.00000272 kilowatt hour
- 0.0093 B.T.U.

1 CHEVAL-VAPEUR =

- 75 kilogrammeters per second
- 0.9863 horse power
- 0.7357 kilowatt

APPENDIX F.

MISCELLANEOUS CONVERSION FACTORS.

1 Pound per Square Inch =

- 2.0355 inches of mercury at 32° F.
- 2.0416 inches of mercury at 62° F.
- 2.309 feet of water at 62° F.
- 0.07031 kilogram per square centimeter
- 0.06804 atmosphere
- 51.7 millimeters of mercury at 32° F.

1 Foot of Water at 62° F. =

- 0.433 pound per square inch
- 62.355 pounds per square foot
- 0.883 inch of mercury at 62° F.
- 821.2 feet of air at 62° F. and barometer 29.92

1 Inch of Water 62° F. =

- 0.0361 pound per square inch
- 5.196 pounds per square foot
- 0.5776 ounce per square inch
- 0.0736 inch of mercury at 62° F.
- 68.44 feet of air at 62° F. and barometer 29.92

1 Foot of Air at 32° F. and Barometer 29.92 =

- 0.0761 pound per square foot
- 0.0146 inch of water at 62° F.

1 Inch of Mercury at 62° F. =

- 0.4912 pound per square inch
- 1.132 feet of water at 62° F.
- 13.58 inches of water at 62° F.

1 Atmosphere =

- 760.0 millimeters of mercury at 32° F.
- 14.7 pounds per square inch
- 29.921 inches of mercury at 32° F.
- 2,116.0 pounds per square foot
- 1.033 kilograms per square centimeter

1 Millimeter = 0.03937 inch

1 Centimeter = 0.3937 inch

1 Meter = 39.37 inches

1 Meter = 3.2808 feet

1 Square Meter = 10.764 square feet

1 Liter =

- 61.023 cubic inches
- 0.264 U. S. gallons

1 Gram =

- 1 cubic centimeter of distilled water
- 15.43 grains troy
- 0.0353 ounce

1 Kilogram =

- 2.20462 pounds avoirdupois

APPENDIX G.

RULES FOR FIREMEN USING ILLINOIS AND INDIANA COAL IN HAND-FIRED FURNACES.

(Formulated by the Coal Stoking and Anti-Smoke Committee of the Illinois Coal Operator's Association.)

1. Break all lumps and do not throw any in furnace any larger than one's fist. The reason for this is, that large lumps do not ignite promptly and their presence also causes holes to form in the fire, which allow the passage of too much air.

2. Keep the ash pits bright at all times. If they become dark it is evident that the fire is getting dirty and needs cleaning, which, if not done, will cause imperfect combustion and smoke. If the furnace is equipped with a shaking grate, it should be operated often enough to prevent any accumulation of ashes in the fire. Do not allow ashes to collect in the ash pits, as they not only shut off the air supply, but may cause the grate to be burned.

3. In firing do not land the coal all in one heap, but spread it over as wide a space as possible as it leaves the shovel. A little practice will enable one to catch the proper motion to give the shovel to make the coal spread properly.

4. Place the fresh coal from the bridge wall forward to the dead plate and do not add more than 3 or 4 shovels at a charge. If this amount makes smoke it should be reduced till smoke ceases, which means, of course, that firing will be at more frequent intervals than formerly to keep up steam. This rule applies in cases where the boiler is worked at a large capacity. In such instances, however, where a small capacity only is required, firing by the coking method is the best, wherein the fresh coal is placed at the front of the fire and pushed back and leveled when it has become coked.

5. Fire one side of the furnace at a time so that the other side containing a bright fire will ignite the volatile gases from the fresh charge.

6. Do not allow the fire to burn down dull before charging. If this is done, it will not only result in a smoky chimney, but an irregular steam pressure.

7. Do not allow holes to form in the fire. Should one form, fill it by leveling and not by a scoop full of coal. Keep the fire even and level at all times. As far as possible level the fire after the coal has become coked.

8. Carry as thick a fire as the draft will allow, but in deciding on the proper thickness, judgment must be exercised. If the draft is poor a thin fire will be in order, but if strong, a thicker fire should be carried.

9. Regulate the draft by the bottom or ash pit doors and not by the stack dampers, because when the stack damper is used it tends to produce a smoky chimney, as it reduces the draft, while the closing of the ash pit door diminishes the capacity to burn coal. If strict attention is given to firing, and accounting to demand, for steam, there will be no occasion to have recourse to dampers, except when there is a sudden interruption in the amount of steam being used.

10. A good general rule is to fire little and often, according to steam demands, rather than heavy and seldom. The former means economy in fuel and a clean chimney, while the latter signifies extravagance in fuel and a smoky chimney.

APPENDIX H.

MOLLIER'S DIAGRAM.

The steam tables give values of the simultaneous physical properties of steam, such as pressure, entropy, temperature, etc. When certain of these properties are known the remainder can be obtained from the tables. The simultaneous properties can also be shown by means of a diagram each point on which represents steam in a perfectly definite condition.

Fig. 462 gives a skeleton outline of such a diagram and Fig. 463 a

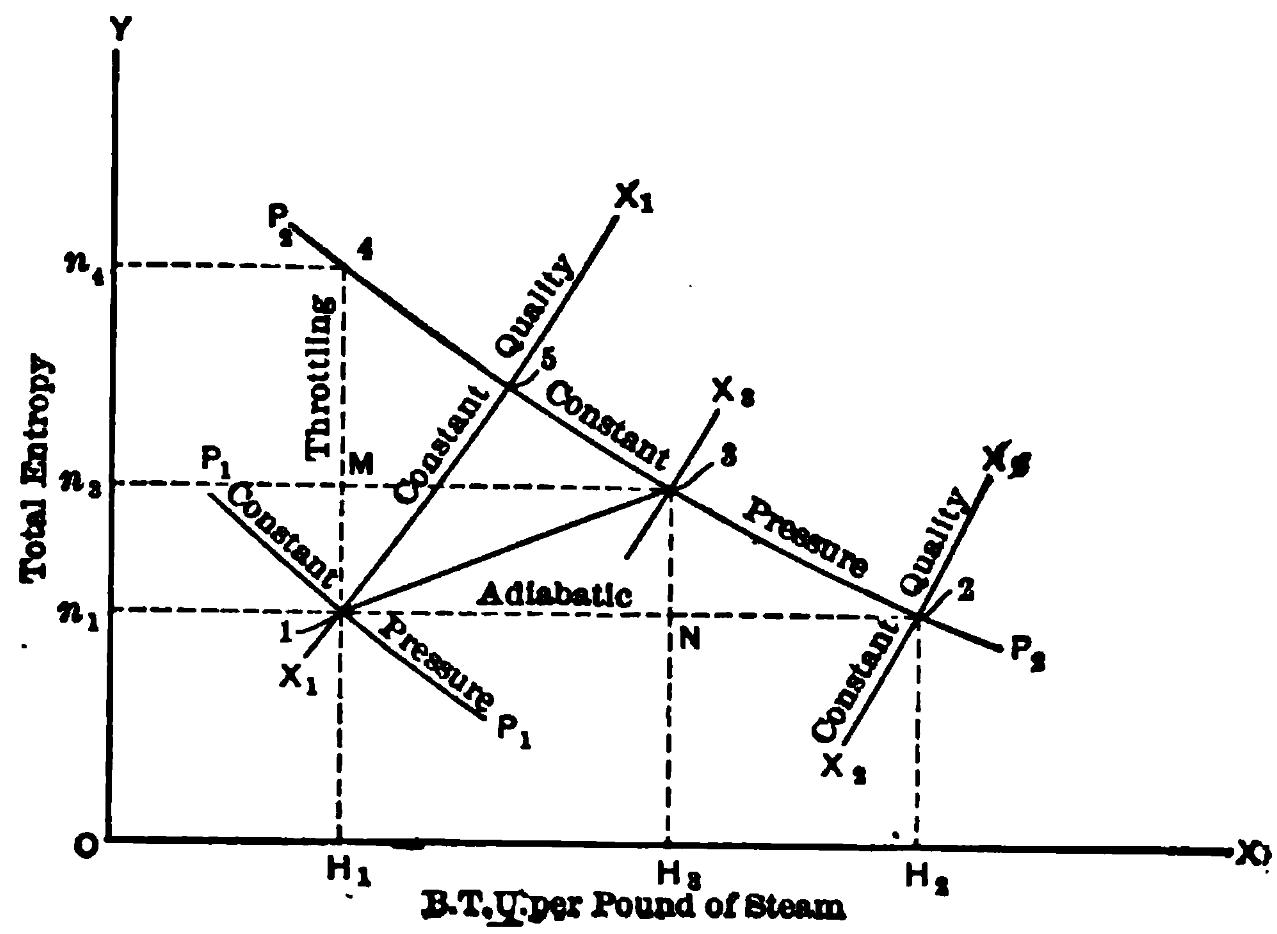


FIG. 462.

reduced reproduction of the complete chart as ordinarily constructed. Referring to Fig. 462, abscissas represent the heat contents or B.T.U. per pound of steam and ordinates represent the total entropy. Vertical lines then represent lines of constant heat content, and horizontal lines constant entropy. P_1P_1 and P_2P_2 represent lines of constant pressure and X_1X_1 and X_2X_2 lines of constant quality. Evidently any point in the chart represents a fixed condition of heat content, pressure, quality and entropy as determined by its location with respect to the different lines. Thus point 1 represents a pressure P_1 as determined by the numerical value of line P_1P_1, quality x_1 by its location on line X_1X_1,

entropy n_1 by its projection N_1 on the Y axis, and heat content H_1 by its projection on the X axis.

The principal advantages of a total heat-entropy diagram over the tables are that they give the properties of wet and superheated steam and offer a simple means of solving many problems without calculations. For example, the chart offers a ready solution of problems involving

(a) Adiabatic expansion.
(b) Throttling.
(c) Expansion with frictional resistances.

(a) *Adiabatic Expansion:* From thermodynamics we know that during an adiabatic change the entropy is constant; thus, in expanding from pressure P_1 and condition represented in point 1 to a lower pressure P_2 it is only necessary to find the intersection 2 of a horizontal line from point 1 with line P_2P_2. The various properties corresponding to point 2 can be read directly from the diagram.

The line $1\text{–}2 = H_1H_2$ represents the *difference* in heat content following adiabatic expansion from pressure P_1 and condition 1 to pressure P, or

line $$H_1H_2 = H_1 - H_2 = x_1r_1 + q_1 - x_2r_2 - q_2.$$

The quality x_2 is read directly from the intersection of line 1–2 with the constant quality line X_2X_2.

The entropy n_2, of course, remains the same.

From equation (73), p.   we find that the velocity due to adiabatic expansion is

$$V = 223.9\sqrt{H_1 - H_2}.$$

Mollier has added along the margin of the diagram (Fig. 463) a scale of velocity so that V may be ascertained by laying off the length H_1H_2 on the scale.

Example: Steam at 120 pounds absolute, quality 0.98, expands adiabatically to a back pressure of 2 pounds absolute. Find the quality and heat content at the lower pressure.

From Fig. 462 we locate P_1 at the intersection of pressure curve 120 and quality curve 0.98. The corresponding values of H_1 and n_1 are found by interpolation to be 1174.7 and 1.564 repectively. Follow horizontal line 1.564 until it intersects pressure line P_2. The corresponding values of H_2 and x_2 are found to be 910 and 0.797 respectively. The horizontal intercept between the two pressure lines laid off on the velocity diagram gives $V = 3640$ feet per second.

Supposing the steam to be superheated 200 degrees instead of being wet, find the quality and heat content at the end of expansion.

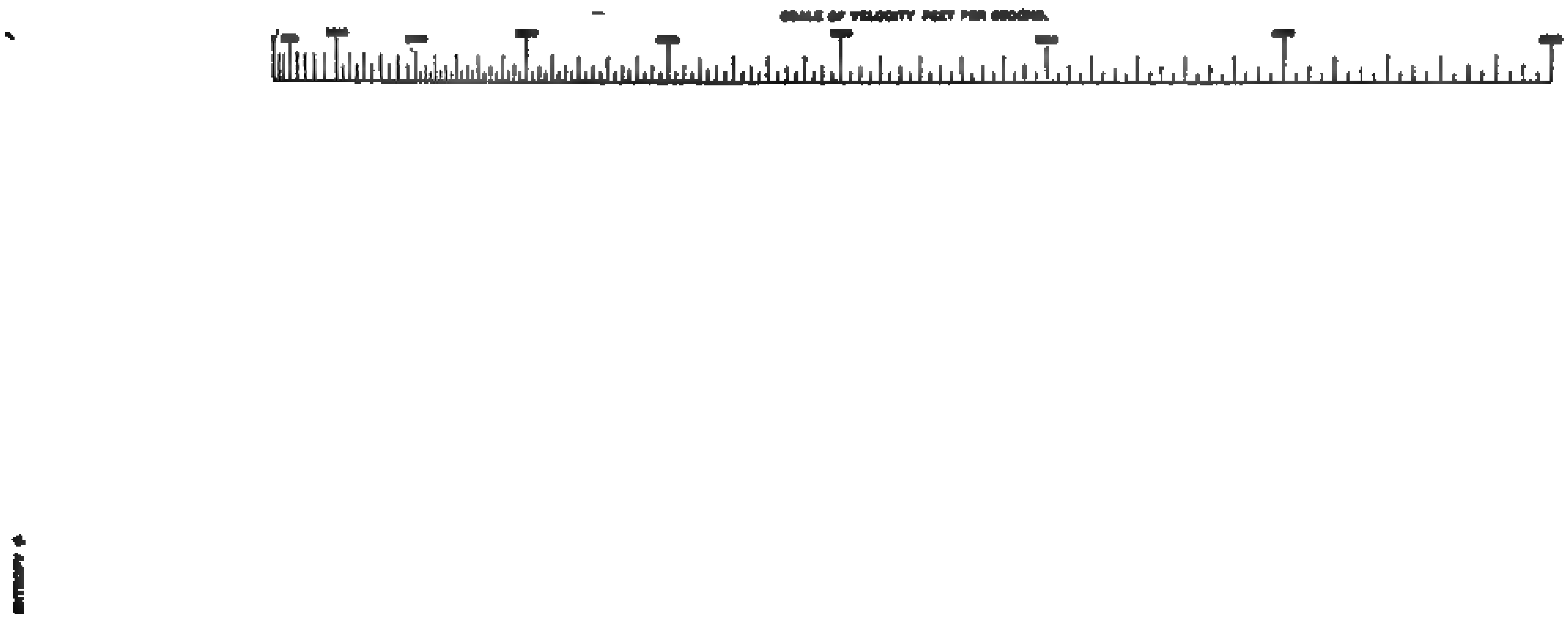


FIG. 463. Total Heat-Entropy Diagram. (Mollier Diagram.)

Locate P_1 at the intersection of pressure curve 120 and superheat curve 200. The corresponding values of H_1 and n_1 are found to be 1295 and 1.703 respectively. Follow horizontal line 1.703 until it intersects pressure line P_2. The corresponding values of H_2 and x_2 are found to be 990 and 877 respectively.

(b) *Throttling:* If steam expands through a small orifice without the addition or abstraction of heat and is brought finally to its initial condition its total heat will be unchanged. This process is called throttling and occurs when steam passes through a reducing valve. Vertical lines in Figs. 462 and 463 are lines of constant total heat and consequently show the changes in the condition of steam which result from throttling. Thus in throttling steam from pressure P_1, Fig. 462, to P_2 it is only necessary to find the intersection, 4, of a *vertical* line from point 1 with line P_2P_2.

Example: Steam at 200 pounds pressure and quality 0.96 passes through a reducing valve and its pressure is lowered to 15 pounds. Find its quality at the lower pressure. The intersection of pressure line 190 with quality line 0.96 gives $H_1 = 1165$. Follow vertical line 1165 until it intersects pressure line 15. The corresponding value for x_2 is found to be 30, that is, the steam is superheated 30 degrees.

To what pressure must the steam be reduced in order that it may be dry and saturated? Follow vertical line 1165 until it intersects the *saturation* curve. The corresponding pressure is found to be 30 pounds.

(c) *Expansion Involving Frictional Resistances:* As steam expands in the nozzle of a turbine or passes between the vanes it experiences frictional resistances which cause it to give up less energy than it would under ideal conditions. The work of friction causes the entropy of the steam at its lowest temperature to be greater than it would be if adiabatic expansion occurred and serves to increase its dryness fraction.

If y one hundredths of the heat $H_1 - H_2$ (given up in adiabatic expansion) is lost due to friction, the heat available for useful work is

$$(1 - y)(H_1 - H_2),$$

the resulting velocity of the jet is

$$V = 223.9\sqrt{(1 - y)(H_1 - H_2)},$$

and the increase in quality of the exhaust steam is

$$\frac{y(H_1 - H_2)}{r_2}.$$

These equations may be readily solved by means of the diagram.

Referring to Fig. 462 line 1 3 represents an expansion from pressure P_1 to pressure P_2 with frictional resistances.

From the diagram

$$y = \frac{N\,2}{12} = \frac{H_3H_2}{H_2H_1}.$$

$$(1 - y)\,(H_1 - H_2) = \text{line } 1\,N = H_1H_3.$$

Increase in quality $= \dfrac{y\,(H_1 - H_2)}{r_2}$ = distance 2 3 between X_2X_2 and X_3X_3.

Increase in entropy $= N\,3 = N_1N_3$.

Example: Steam at 160 pounds absolute initial pressure, quality 0.97, expands through a nozzle to a back pressure of 2 pounds absolute. If 15 per cent of the heat energy is lost in friction, find the quality of the steam at the lower pressure and the velocity of the jet.

From Fig. 462 we locate P_1 at the intersection of pressure curve 160 and quality curve 0.97. The corresponding value of H_1 is 1170. Follow line 1170 horizontally until it intersects pressure line P_2. From the diagram we find for adiabatic expansion $H_2 = 910$ $x_2 = 0.797$. But the friction increases the heat content at the end of expansion an amount $0.15 \times H_1 - H_2 = 0.15\,(1170 - 910) = 39$, so that the final heat content $= 910 + 39 = 949$.

Follow pressure line P_2 until it intersects heat line 949. The quality x_2 is found to be 0.836 and the entropy $n_2 = 1.632$. From the velocity scale we find $V = 3320$ feet per second for $H_1 - H_2 = (1170 - 949)$.

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